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TECHNICAL NOTE 2025

EFFECT OF VALVE OVERLAP AND COMPRESSION RATIO ON VARIATION
OF MEASURED PERFORMANCE WITH EXHAUST PRESSURE OF
AIRCRAFT CYLINDER AND ON COMPUTED PERFORMANCE
OF COMPOUND POWER PLANT

By Carroll S. Eian

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SUMMARY

An investigation was conducted on an aircraft liquid-cooled single-cylinder test engine (four-stroke-cycle spark-ignition type) equipped with direct fuel injection to determine the effect of valve overlap and compression ratio on variation of engine power and specific fuel consumption with exhaust pressure. At engine speeds of 2000 and 2600 rpm, runs were made for compression ratios of 5.5 and 7.6 each at valve overlaps of 63° , 166° , and 200° over ranges of exhaust pressure from 10 to 60 inches of mercury absolute and of fuel-air ratio from 0.035 to 0.100. Inlet-manifold pressures were 30, 40, and 50 inches of mercury absolute.

Increasing valve overlap from 63° to 200° changed the variation of power with increasing exhaust pressure from a slight decrease to an increase of about the same magnitude when the exhaust pressure was less than about 80 percent of the manifold pressure. At higher exhaust pressures, the rate of decrease in power with exhaust pressure became so great with the larger valve overlaps (166° and 200°) that the engine could not be operated when the exhaust pressure was greater than about 95 percent of the inlet-manifold pressure. Indicated specific fuel consumption tended to increase with increased valve overlap. Changing engine compression ratio from 5.5 to 7.6 and engine speed from 2000 to 2600 rpm did not greatly change the general effect of valve overlap and exhaust pressure on performance.

From data of this investigation, performance calculations were made for a compound power plant assuming a system in which a steady-flow turbine and compressor are mounted on the same shaft and the turbine power not required for supercharging is returned to the engine through gears. When the valve overlap was changed from 63° to 200° at an engine speed of 2000 rpm, maximum net brake specific power and the

corresponding net specific fuel consumption were increased approximately 17 and 30 percent, respectively. At an engine speed of 2600 rpm when valve overlap was changed from 63° to 150° , the maximum net brake specific power of the compound power plant was increased about 7 percent, whereas the net specific fuel consumption remained nearly constant. With further increase to 200° valve overlap, the net brake specific power decreased with an accompanying increase in fuel consumption.

INTRODUCTION

Analysis of the compound power plant consisting of a conventional four-stroke-cycle spark-ignition reciprocating engine in combination with a gas turbine and a compressor is generally based on the use of current reciprocating-engine designs for which the valve overlap averages about 50° . Preliminary analysis of the effect on compound-power-plant performance of increasing valve overlap to values considerably above those currently used indicated that the charge-air flow could be substantially increased resulting in better cylinder scavenging with consequent increase in power and, in addition, an increase in the net power available to the crankshaft from the turbine and the compressor components.

Data obtained on an air-cooled aircraft cylinder indicated an increase in engine power of about 20 percent for a change in valve overlap from 40° to 130° (reference 1). The effect of changing valve overlap from 40° to 62° on compound-power-plant performance is given in reference 2.

In order to determine whether an optimum valve overlap exists, particularly when the engine is considered as a component of a compound-power-plant, an investigation reported herein was conducted at the NACA Lewis laboratory on an aircraft-type liquid-cooled single-cylinder test engine with three valve overlaps (63° , 166° , and 200°) for various engine speeds, inlet-manifold pressures, exhaust pressures, and fuel-air ratios. The current design values of compression ratio were bracketed by investigating the effect of valve overlap and exhaust pressure at compression ratios of 5.5 and 7.6

The effect of the foregoing variables on measured engine power and specific fuel consumption are presented together with their effect on the calculated performance of a compound power plant.

APPARATUS

An aircraft-type liquid-cooled cylinder with a $5\frac{1}{4}$ -inch bore and $4\frac{3}{4}$ -inch stroke, incorporating single intake and exhaust valves, was mounted on a single-cylinder-engine test stand equipped with standard accessories for determining engine speed and torque. A schematic diagram of the single-cylinder engine and associated equipment is shown in figure 1. Compression ratios of 5.5 and 7.6, which were obtained by placing steel plates of predetermined thickness between the cylinder and the crankcase, were experimentally checked.

The valve-overlap period was varied by changing the time of exhaust-valve closing through the use of a two-piece overhead camshaft with an individual exhaust cam for each valve overlap investigated. The period of dwell of the exhaust event was therefore the only difference between cam profiles for each valve overlap. The timing of the other valve events was not reproduced exactly for the three different valve overlaps because of small changes in valve timing that were introduced by mechanical difficulties. Valve events for the three overlaps are shown in the following table:

Valve timing (deg)	Valve overlap (deg)		
	63	166	200
Intake opens, B.T.C.	33	32	30
Intake closes, A.B.C.	37	40	46
Exhaust opens, B.T.C.	57	57	58
Exhaust closes, A.T.C.	30	134	170

A valve-lift diagram for 166° overlap is given in figure 2.

Fuel was introduced into the cylinder by direct injection into the head between the valves through a commercial spring-loaded injection nozzle; the spray cone angle was about 60° . Fuel was supplied to the nozzle by a plunger-type pump incorporating a variable stroke to control the rate of fuel delivery. The pump was mechanically driven by the engine and was equipped with an adjustable coupling for changing the time of fuel injection. The fuel (AN-F-28, Amendment-2) flow was measured by a calibrated rotameter with provisions for damping the intermittent flow.

Combustion air was metered to the cylinder by a flange-tap thin-plate orifice installed according to A.S.M.E. specifications. Automatically controlled electric heaters were used to regulate the

inlet-manifold temperature. Exhaust pressure was controlled by a butterfly valve in the laboratory exhaust system downstream of an exhaust-gas mixing tank located at the exhaust of the cylinder. The cylindrical exhaust mixing tank had a tangential entry, a volume about 23 times that of the cylinder volume, and was covered with 2 inches of thermal insulation. Exhaust pressure was measured in the tank about 2 inches below the entry of the engine exhaust pipe. Exhaust temperature was measured by two shielded thermocouples located in the exit of the tank.

PROCEDURE

Operating conditions. - The range of operating conditions for which data were obtained is listed in table I.

Data were obtained with valve overlaps of 63° , 166° , and 200° at each of the two compression ratios, 5.5 and 7.6. Each combination of valve overlap and compression ratio was investigated at several inlet-manifold pressures for engine speeds of 2000 and 2600 rpm. For each combination of valve overlap, compression ratio, inlet-manifold pressure, and engine speed, the over-all fuel-air ratio was varied through a range of approximately 0.035 to 0.080 at several constant values of the ratio of exhaust pressure to inlet-manifold pressure p_e/p_m . At an over-all fuel-air ratio of 0.065, several runs were also made with variable exhaust pressure.

In general, the inlet-manifold temperature was maintained constant at 200°F ; however, a few runs were made for which the manifold temperature was varied in order to determine the effect on power.

The oil-inlet temperature was held at $160^\circ \pm 5^\circ\text{F}$. The average of the coolant temperatures at the entrance and exit of the cylinder was held at 175°F ; the maximum coolant temperature rise was about 15°F for the range of conditions. The coolant flow rate was maintained at 125 ± 2 pounds per minute and the coolant-system pressure was about 23 ± 3 pounds per square inch gage.

Injection timing. - With large valve overlaps a considerable amount of charge air can, for certain operating conditions, pass completely through the engine without being trapped in the cylinder during the compression and expansion strokes. For such conditions, if fuel is injected directly into the cylinder a considerable difference may exist between the cylinder fuel-air ratio and the over-all fuel-air ratio. In addition, when the exhaust valve closes very late, as in this investigation of the larger valve overlaps

(166° and 200°), part of the fuel may possibly be blown through the cylinder during the scavenging process if fuel injection begins too early. Early injection, however, is desirable for good mixing of fuel and air.

In order to determine the optimum time for fuel injection with the larger overlaps, a preliminary investigation was made in which the time of injection was made progressively earlier with all other operating conditions constant. The point at which engine power began to decrease was noted and the time of injection was then set to begin a few degrees before loss of power. This preliminary investigation was made at a p_e/p_m of 0.75 and an over-all fuel-air ratio of 0.065. The injection timings obtained, which were used throughout the remainder of the investigation, are included in the following table together with the timing for 63° overlap that was arbitrarily set to begin a few degrees after exhaust-valve closure:

Valve overlap (deg)	Beginning of injection (deg A.T.C.)
63	44 (14° after exhaust valve closes)
166	55 (79° before exhaust valve closes)
200	90 (80° before exhaust valve closes)

Frequent checks were made throughout the entire investigation to insure consistent performance of the fuel injection valve inasmuch as preliminary tests indicated the need of attention to prevent erratic changes in combustion efficiency due to inconsistent operation of the injection valve.

Motoring runs. - Motoring runs were made at a p_e/p_m of 1.0 for all combinations of valve overlap, compression ratio, inlet-manifold pressure, and engine speed in order to estimate rubbing friction power. The resulting values of friction horsepower and corresponding operating conditions are listed in table II.

RESULTS AND DISCUSSION

Experimental Engine Data

In the study of the effect of valve overlap and compression ratio on the variation of measured aircraft-engine cylinder performance with exhaust pressure, the following engine performance parameters were considered:

Indicated power. - As in references 3 to 6, the dimensionless quantity ϕ (ratio of indicated mean effective pressure to inlet-manifold pressure) is used as a measure of engine power. The quantity ϕ is computed from the experimental data by the relation

$$\phi = \frac{\text{imep}}{p_m} = \frac{2 \times 33,000 \text{ ihp}}{p_m v_d N}$$

where

imep indicated mean effective pressure of engine, (lb/sq ft)

ihp indicated horsepower of engine

p_m inlet-manifold pressure, (lb/sq ft absolute)

v_d displacement volume of engine, (cu ft)

N engine speed, (rpm)

The indicated horsepower was taken as the sum of the brake horsepower and the appropriate value of friction horsepower from table II. Inasmuch as friction horsepower was obtained at a p_e/p_m of 1.0, the pumping power during the motoring runs can be considered zero and hence the indicated horsepower, as defined herein, can be considered to include the contributions of all four strokes of the cycle.

In figure 3, ϕ is plotted against p_e/p_m for constant engine speed, valve overlap, and compression ratio. A single curve is obtained (as in references 4 to 6) regardless of whether p_e or p_m is varied, provided that the inlet-manifold temperature is constant. Changes in engine speed and compression ratio do not have much effect on the general shape of the curves but do affect the magnitude of ϕ . At constant valve overlap and p_e/p_m , ϕ is generally slightly greater at the lower engine speed and at the higher compression ratio. The increase in ϕ with increased compression ratio is slightly less than would be expected from the change in thermal efficiency, because the volumetric efficiency decreased slightly with increased compression ratio.

Increasing the valve overlap had an appreciable effect on the variation of ϕ with p_e/p_m . With 63° overlap (fig. 3), the curves have the characteristic shape usually found with conventional values

of overlap (references 4 to 6); that is, as p_e/p_m increases, ϕ progressively decreases, the rate of decrease becoming somewhat greater for values of p_e/p_m above about 0.7. The inflection point for values of p_e/p_m of about 1.1 was also obtained in investigations on multicylinder engines having comparable valve overlaps (references 5 and 6). Investigations with lower valve overlaps (that is, 40°) did not always show this inflection point (references 1, 2, and 4). Increasing the valve overlap to 166° resulted in a somewhat smaller rate of decrease of ϕ with increase in p_e/p_m for values of p_e/p_m below about 0.8; however, above 0.8 the rate of decrease was much greater than with 63° overlap. In fact, power (and hence ϕ) decreased so rapidly that the engine could not be operated when p_e/p_m was greater than about 0.950.

With 200° overlap, extremely rapid decrease of ϕ as p_e/p_m was increased above 0.8 is again evident. With this overlap, however, ϕ also decreased with a decrease in p_e/p_m below 0.8. A similar result is recorded in reference 1 with a valve overlap of 130° and is attributed to the large scavenging air flow carrying an appreciable percentage of the heat of combustion to the exhaust at low values of p_e/p_m .

An additional contributing factor to this effect is the change in cylinder fuel-air ratio (in spite of constant over-all fuel-air ratio) accompanying changes in p_e/p_m . Inasmuch as cylinder injection was used, practically all the fuel can be considered to be trapped in the cylinder. The trapped air flow, however, becomes a smaller percentage of the total as p_e/p_m is decreased from the value for which no blow-through exists, resulting in an increase in cylinder fuel-air ratio beyond the maximum power value (that is, 0.08). With large valve overlap, a low p_e may also be expected to cause a reduction in the pressure of the air trapped in the cylinder and hence in ϕ . The foregoing effects are opposed by the decreased residual gas and the decreased pumping losses (increased pumping recovery) accompanying a decrease in exhaust pressure.

The effect of valve overlap on power is further illustrated in figure 4 where ϕ is plotted against valve overlap for several values of p_e/p_m at a compression ratio of 5.5 and engine speeds of 2000 and 2600 rpm. Similar trends were obtained for the same engine speeds at a compression ratio of 7.6. Although a high degree of accuracy cannot be claimed for the middle region of the curves

because of lack of experimental data, the curves indicate that maximum power is obtained at a valve overlap of about 120° for the lower values of p_e/p_m (0.333 and 0.667). The data of reference 1 also show an increase in power of about 12 percent for an increase in valve overlap from 40° to 130° at similar values of p_e/p_m and an engine speed of 2000 rpm, which compares favorably with the corresponding improvement indicated by the curves in figure 4(a) if they are extrapolated to a valve overlap of 40° . At p_e/p_m of 0.833, the variation of power with valve overlap is small at both engine speeds. At p_e/p_m of 0.950, the effect of valve overlap is negligible at 2000 rpm, but at 2600 rpm there is a continual decrease of ϕ with increasing overlap over the range investigated.

The variation of ϕ with over-all fuel-air ratio for all engine conditions investigated is shown in figure 5. These plots further illustrate the effect of charge-air blow-through in changing the cylinder fuel-air ratio when valve overlap and p_e/p_m are varied. With a valve overlap of 63° , maximum power is reached at practically all combinations of p_e/p_m , engine speed, and inlet-manifold pressure at an over-all fuel-air ratio of 0.075 to 0.080, which agrees with that of a carbureted engine. With 200° overlap, however, maximum power is reached at an over-all fuel-air ratio as low as 0.055 at a p_e/p_m of 0.333 where charge-air blow-through is large and cylinder fuel-air ratio is probably about 0.08 (maximum power value).

Volumetric efficiency. - Volumetric efficiency η_v , defined as the ratio of the volume of charge air taken into the engine per cycle at inlet-manifold pressure and temperature to the displacement volume of the engine, is calculated from the relation

$$\eta_v = \frac{2W_c R(T_m + 460)}{p_m v_d N}$$

where

W_c charge-air flow, (lb/min)

R gas constant for air, (ft-lb/(lb)(°F))

T_m inlet-manifold temperature, (°F)

The variation of η_v with p_e/p_m is shown in figure 6 for all conditions investigated at an over-all fuel-air ratio of 0.065.

With other conditions constant, a single curve of η_v against p_e/p_m is obtained regardless of whether p_e or p_m is varied. The general shape of the curves for a constant overlap is not greatly affected by changes in either engine speed or compression ratio; however, η_v is generally slightly greater at the lower engine speed and compression ratio.

Increasing the valve overlap has a considerable effect on the variation of η_v with p_e/p_m . For example, at 2000 rpm and a compression ratio of 5.5, a decrease in p_e/p_m from 0.95 to 0.30 results in an increase in η_v of about 14 percent with 63° overlap, of 45 percent with 166° overlap, and of 70 percent with 200° overlap. Equally large effects are found for the other combinations of engine speed and compression ratio. Inspection of the figure also shows that at compression ratio of 5.5, a change in valve overlap from 63° to 200° increases the value of η_v at p_e/p_m of 0.95 about 6 percent at 2000 rpm and decreases η_v about 28 percent at 2600 rpm.

Indicated specific fuel consumption. - The variation of indicated specific fuel consumption with over-all fuel-air ratio is presented in figure 7 for all engine conditions investigated. The effects of the variation of cylinder fuel-air ratio with charge-air blow-through are again apparent. For example, with 63° valve overlap, where the blow-through is small, the minimum indicated specific fuel consumption occurs at an over-all fuel-air ratio between 0.055 and 0.065, which corresponds to values for a carbureted engine. With 200° valve overlap, however, at p_e/p_m of 0.333, where the charge-air blow-through is large, the minimum indicated specific fuel consumption occurs at an over-all fuel-air ratio of 0.040 to 0.050. At 200° overlap, as p_e/p_m is increased to 0.950, the over-all fuel-air ratio for minimum specific fuel consumption increases to a value corresponding to that of a carbureted engine (0.055 to 0.065). This variation is caused by the decrease in blow-through of the charge air with increasing exhaust pressure and indicates that the blow-through is small at a value of p_e/p_m of 0.950.

The curves also show that for otherwise constant conditions minimum specific fuel consumption tends to increase as valve overlap is increased over the range investigated. Specific fuel consumption is slightly increased by a change in engine speed from 2000 to 2600 rpm. Increasing the compression ratio from 5.5 to 7.6 results in a decrease in fuel consumption as would be expected because of the improvement in thermal efficiency of the engine. At constant p_e/p_m , the variation of specific fuel consumption with over-all fuel-air ratio is substantially independent of inlet-manifold pressure.

Variation of power with charge-air flow. - For a given fuel-air ratio when no blow-through occurs, the power of the compression and expansion strokes is proportional to charge-air flow. An estimate of the horsepower of the compression and expansion strokes of the cycle is the indicated horsepower (previously defined) minus the pumping horsepower, $ihp - php$. An approximation of the pumping horsepower php is obtained from the following equation:

$$php = \frac{(p_m - p_e) v_d N}{2 \times 33,000}$$

where

p_e exhaust pressure, (lb/sq ft absolute)

This relation for pumping horsepower assumes a square indicator card for the intake and exhaust strokes. In figure 8, $ihp - php$ is plotted against charge-air flow W_c for all the engine conditions investigated at an over-all fuel-air ratio of 0.065. Separate plots are shown for each of the two compression ratios. The data for 63° overlap fall on a straight line passing through the origin, thus indicating proportionality between power and air flow. With the larger overlaps, however, as the air flow is increased by decreasing p_e/p_m , the power does not increase proportionately but tends to attain a limiting value, indicating that an increasing amount of air is passing through the engine without being trapped in the cylinder.

As would be expected, the slope of the line, which is an indication of thermal efficiency, is greater for a compression ratio of 7.6 than for 5.5.

Variation of power with inlet-manifold temperature. - The plot in figure 9 shows the variation of ϕ with over-all fuel-air ratio to be similar for the different inlet-manifold temperatures over the range shown. These data, which were of an exploratory nature obtained from an early version of the setup, are not exactly comparable with the data of figure 3(b) because of differences in the methods of exhaust-gas pressure measurement.

A cross plot of figure 9 is given in figure 10 in which the ratio of ϕ at any inlet-manifold temperature to ϕ at an inlet-manifold temperature of 200° F $\phi_{T_m}/\phi_{200^\circ}$ is plotted against inlet-manifold temperature for a range of over-all fuel-air ratios. The variation of ϕ with inlet-manifold temperature is approximately inversely proportional to the first power of the absolute inlet-manifold temperature.

Exhaust-gas temperature. - Typical curves showing the variation of measured exhaust-gas temperature with over-all fuel-air ratio are presented in figure 11 for valve overlaps of 63°, 166°, and 200° at several values of p_e/p_m . These plots are for an engine speed of 2600 rpm and an inlet-manifold pressure of 40 inches of mercury absolute for compression ratios of 5.5 and 7.6. At 63° valve overlap, the curves are fairly flat but generally reach a maximum value at an over-all fuel-air ratio of about 0.07. However, at the two higher valve overlaps and at low values of p_e/p_m where considerable blow-through of charge air occurs, the exhaust-gas temperature increases rapidly with an increase in over-all fuel-air ratio from about 0.050 to 0.070, indicating afterburning of excess fuel and charge air in the exhaust-gas mixing tank. Exhaust-gas temperatures generally increase with a decrease in p_e/p_m and at a greater rate for valve overlaps of 166° and 200° than for 63°.

Compound-Power-Plant Performance

The calculations for the compound power plant using the data obtained in this investigation were made assuming a system in which a steady-flow turbine and a compressor are mounted on the same shaft and the turbine power not required for supercharging is returned to the engine through gears. The net power of such a system was obtained by use of the following equation:

$$\text{net hp} = \text{ihp} - \text{fhp} + \eta_g (\text{thp} - \text{shp})$$

where

net hp	net horsepower of compound system
ihp	indicated horsepower from experimental data
fhp	calculated friction horsepower, $9.33 \times 10^{-7} N^2$
thp	turbine horsepower, power obtained from expanding engine exhaust gas from engine exhaust pressure and temperature to altitude pressure through turbine with adiabatic efficiency of 80 percent
shp	supercharger horsepower, power required to compress charge air from altitude temperature and pressure to inlet-manifold pressure of engine with duct allowance using an efficiency of 80 percent

η_g gear efficiency, assumed to be 95 percent

An aftercooler was assumed to be placed between the supercharger outlet and the inlet manifold of the engine to cool the charge air to 200° F at the inlet manifold. Values of aftercooler effectiveness necessary to accomplish this cooling are under 30 percent for all the conditions investigated. Turbine power was calculated by the use of data from reference 7. The calculations are for an altitude of 30,000 feet, engine speeds of 2000 and 2600 rpm, inlet-manifold pressure of 40 inches of mercury absolute, and compression ratios of 5.5 and 7.6.

Plots were made of the calculated net brake specific power of the compound system (net hp/cu in.) and corresponding net specific fuel consumption (net sfc) as a function of p_e/p_m for all the over-all fuel-air ratios investigated. From these plots, the net brake specific power corresponding to p_e/p_m and over-all fuel-air ratio for maximum power was determined and is plotted with the corresponding net specific fuel consumption against valve overlap in figure 12. The minimum net specific fuel consumption was determined in the same manner and is plotted with the corresponding net brake specific power against valve overlap. Maximum net brake specific power occurred generally at a p_e/p_m of about 0.7, whereas minimum net specific fuel consumption occurred at a p_e/p_m of about 0.8.

At an engine speed of 2000 rpm (fig. 12(a)), maximum net brake specific power for both compression ratios increases about 17 percent as valve overlap is increased from 63° to 200°. This increase in power is accompanied by an increase in the corresponding net specific fuel consumption of about 30 percent for a compression ratio of 5.5, but at a compression ratio of 7.6, net specific fuel consumption remains nearly constant. Minimum net specific fuel consumption is substantially constant over the range of valve overlaps investigated for both compression ratios, whereas the corresponding net brake specific power for a compression ratio of 5.5 increases over the entire range of valve overlap and is 12 percent greater at 200° than that at 63° overlap. The net brake specific power for a compression ratio of 7.6 reaches a maximum value at about 130° valve overlap, which is about 5 percent greater than that at 63° valve overlap.

At an engine speed of 2600 rpm (fig. 12(b)) when valve overlap is increased from 63° to 150°, net brake specific power for both maximum power and for minimum net specific fuel consumption at both compression ratios increases 6 to 8 percent and the corresponding net specific fuel consumptions are nearly constant. As valve overlap is

increased above 150° , the net brake specific power decreases with accompanying increase in fuel consumption. As would be expected, better performance is obtained at a compression ratio of 7.6 than at 5.5 inasmuch as these curves are for constant inlet-manifold pressure.

The trends of the curves in figure 12 are also representative of results obtained at an inlet-manifold pressure of 50 inches of mercury absolute.

SUMMARY OF RESULTS

The results of an investigation of the effect of valve overlap and compression ratio on variation of measured aircraft-engine cylinder performance with exhaust pressure may be summarized as follows:

1. Increasing valve overlap from 63° to 200° changed the variation of power with increasing exhaust pressure from a slight decrease to an increase of about the same magnitude when the exhaust pressure was less than about 80 percent of the manifold pressure. At higher exhaust pressures, the rate of decrease in power with exhaust pressure became so great with the larger valve overlaps (166° and 200°) that the engine could not be operated when the exhaust pressure was greater than about 95 percent of the inlet-manifold pressure.

2. At low exhaust pressures (33 percent of inlet-manifold pressure), the data indicated that maximum power would be obtained with a valve overlap of about 120° for the compression ratios and engine speeds investigated; at an exhaust pressure equal to 95 percent of inlet-manifold pressure, valve overlap had no appreciable effect on the power obtained at 2000 rpm, but at 2600 rpm the power decreased with an increase in valve overlap over the range investigated for both compression ratios.

3. The minimum indicated specific fuel consumption increased with increase in valve overlap over the range investigated.

4. Changing engine compression ratio from 5.5 to 7.6 and engine speed from 2000 to 2600 rpm did not greatly change the general effect of exhaust pressure on performance.

The results of calculations using the engine data to determine the performance of a compound power plant assuming a system in which

a steady-flow turbine and a compressor are mounted on the same shaft and the turbine power not required for supercharging is returned to the engine through gears showed that:

5. At an engine speed of 2000 rpm, net brake specific power corresponding to ratio of exhaust pressure to inlet-manifold pressure and over-all fuel-air ratio for maximum power increased about 17 percent as valve overlap was increased from 63° to 200° at both compression ratios. The corresponding net specific fuel consumption increased about 30 percent at a compression ratio of 5.5 and remained nearly constant at a compression ratio of 7.6.

6. At an engine speed of 2600 rpm when valve overlap was increased from 63° to 150° , the maximum net brake specific power for both compression ratios increased 6 to 8 percent and the net specific fuel consumption remained nearly constant. As valve overlap was further increased from 150° to 200° , net brake specific power decreased with accompanying increase in fuel consumption.

7. For compression ratios of 5.5 and 7.6 with all other conditions constant, better compound-power-plant performance was generally obtained at the higher compression ratio.

Lewis Flight Propulsion Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio, February 9, 1949.

REFERENCES

1. Creagh, John W. R., Hartmann, Melvin J., and Arthur, W. Lewis, Jr.: An Investigation of Valve-Overlap Scavenging over a Wide Range of Inlet and Exhaust Pressures. NACA TN 1475, 1947.
2. Boman, David S., and Kaufman, Samuel J.: Effect of Reducing Valve Overlap on Engine and Compound-Power-Plant Performance. NACA TN 1612, 1948.
3. Pinkel, Benjamin: Effect of Exhaust Back Pressure on Engine Power. NACA CB 3F17, 1943.
4. Boman, David S., Nagey, Tibor F., and Doyle, Ronald B.: Effect of Exhaust Pressure on the Performance of an 18-Cylinder Air-Cooled Radial Engine with a Valve Overlap of 40° . NACA TN 1220, 1947.

5. Humble, Leroy V., Nagey, Tibor F., and Boman, David S.: Effect of Exhaust Pressure on the Performance of an 18-Cylinder Air-Cooled Radial Engine with a Valve Overlap of 62° . NACA TN 1232, 1947.
6. Desmon, Leland G., and Doyle, Ronald B.: Effect of Exhaust Pressure on the Performance of a 12-Cylinder Liquid-Cooled Engine. NACA TN 1367, 1947.
7. Pinkel, Benjamin, and Turner, L. Richard: Thermodynamic Data for the Computation of the Performance of Exhaust-Gas Turbines. NACA ARR 4B25, 1944.

TABLE I - OPERATING CONDITIONS

Compression ratio	Valve overlap (deg)	Nominal engine speed (rpm)	Inlet-manifold pressure (in. Hg abs.)	Exhaust pressure (in. Hg abs.)	Over-all fuel-air ratio	Inlet-manifold temperature (°F)
5.5	63	2000	40	13.3 to 60	0.049 to 0.079	200
5.5	63	2000	50	16.7 to 60	.049 to .075	200
5.5	63	2600	40	13.3 to 50	.049 to .080	200
5.5	63	2600	50	16.7 to 60	.049 to .075	200
5.5	166	2000	40	13.3 to 38	.040 to .080	200
5.5	166	2000	50	16.7 to 47.5	.040 to .075	200
5.5	166	2600	40	13.3 to 38	.045 to .083	200
5.5	166	2600	50	16.7 to 47.5	.040 to .075	200
5.5	200	2000	30	10 to 28.5	.035 to .070	200
5.5	200	2000	40	13.3 to 38	.035 to .074	200
5.5	200	2000	50	16.7 to 25	.040 to .080	200
5.5	200	2600	40	13.3 to 38	.040 to .080	200
5.5	200	2600	50	16.7 to 37.5	.045 to .080	200
7.6	63	2000	30	10 to 40	.050 to .080	200
7.6	63	2000	40	10 to 50	.045 to .080	200
7.6	63	2600	30	10 to 50	.055 to .084	200
7.6	63	2600	40	10 to 50	.050 to .080	200
7.6	166	2000	30	10 to 25	.040 to .070	200
7.6	166	2000	40	13.3 to 38	.035 to .070	200
7.6	166	2600	30	10 to 26	.040 to .079	200
7.6	166	2600	40	13.3 to 33.3	.040 to .081	200
7.6	200	2000	30	10 to 28	.040 to .065	200
7.6	200	2000	40	13.3 to 38	.035 to .070	200
7.6	200	2600	30	10 to 28	.045 to .080	200
7.6	200	2600	40	13.3 to 38	.040 to .080	200
7.6	63	2000	30	30	.065 to .100	90
7.6	63	2000	30	30	.060	100
7.6	63	2000	30	30	.060	120
7.6	63	2000	30	30	.060	140
7.6	63	2000	30	30	.065 to .100	150
7.6	63	2000	30	30	.060	160
7.6	63	2000	30	30	.060	180
7.6	63	2000	30	30	.060 to .100	200

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TABLE II - FRICTION HORSEPOWER

Compres- sion ratio	Valve overlap (deg)	Nominal engine speed (rpm)	Inlet- manifold pressure (in. Hg abs.)	Friction horse- power ^a
5.5	63	2000	40	11.9
5.5	63	2000	50	12.6
5.5	63	2600	40	17.9
5.5	63	2600	50	19.0
5.5	166	2000	40	11.2
5.5	166	2000	50	11.5
5.5	166	2600	40	16.3
5.5	166	2600	50	17.1
5.5	200	2000	30	10.7
5.5	200	2000	40	11.2
5.5	200	2000	50	11.1
5.5	200	2600	40	15.3
5.5	200	2600	50	16.3
7.6	63	2000	30	10.3
7.6	63	2000	40	10.8
7.6	63	2600	30	15.1
7.6	63	2600	40	15.9
7.6	166	2000	30	9.7
7.6	166	2000	40	10.4
7.6	166	2600	30	14.2
7.6	166	2600	40	15.2
7.6	200	2000	30	9.8
7.6	200	2000	40	10.1
7.6	200	2600	30	14.5
7.6	200	2600	40	14.9



^aExhaust pressure equal to inlet-manifold pressure.

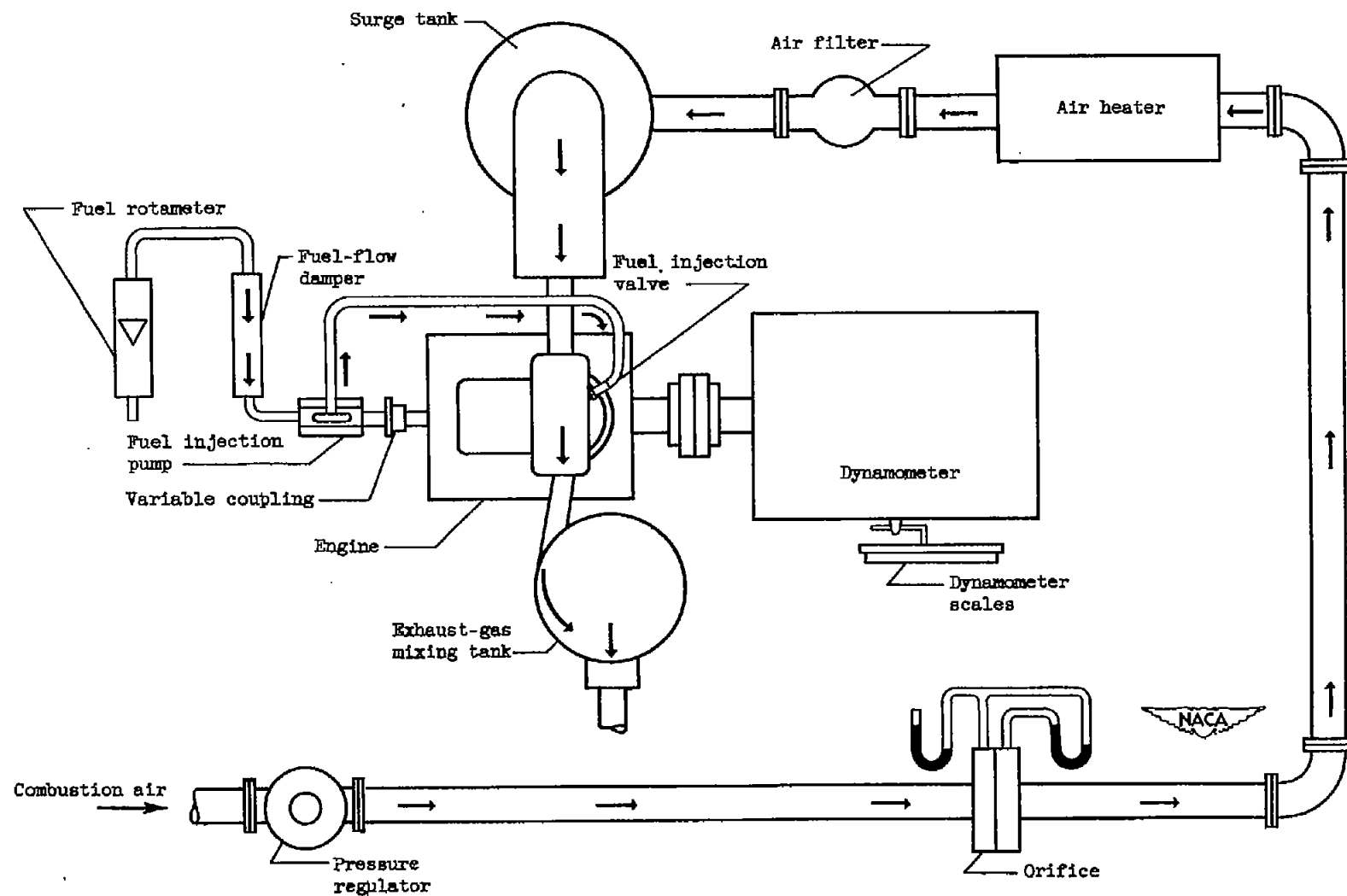


Figure 1. - Schematic diagram of single-cylinder engine and associated equipment.

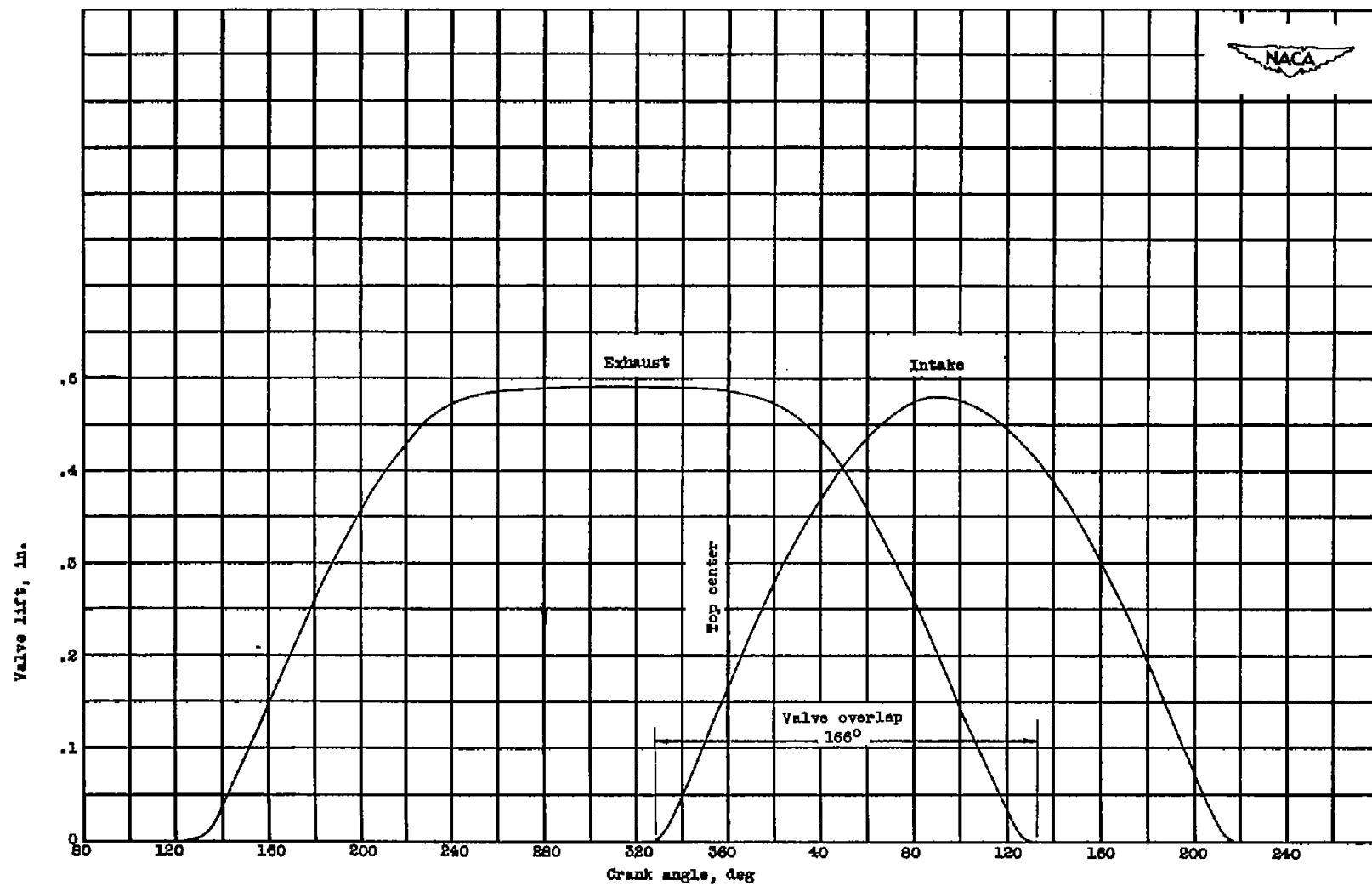


Figure 2. - Valve-lift diagram for valve overlap of 166°.

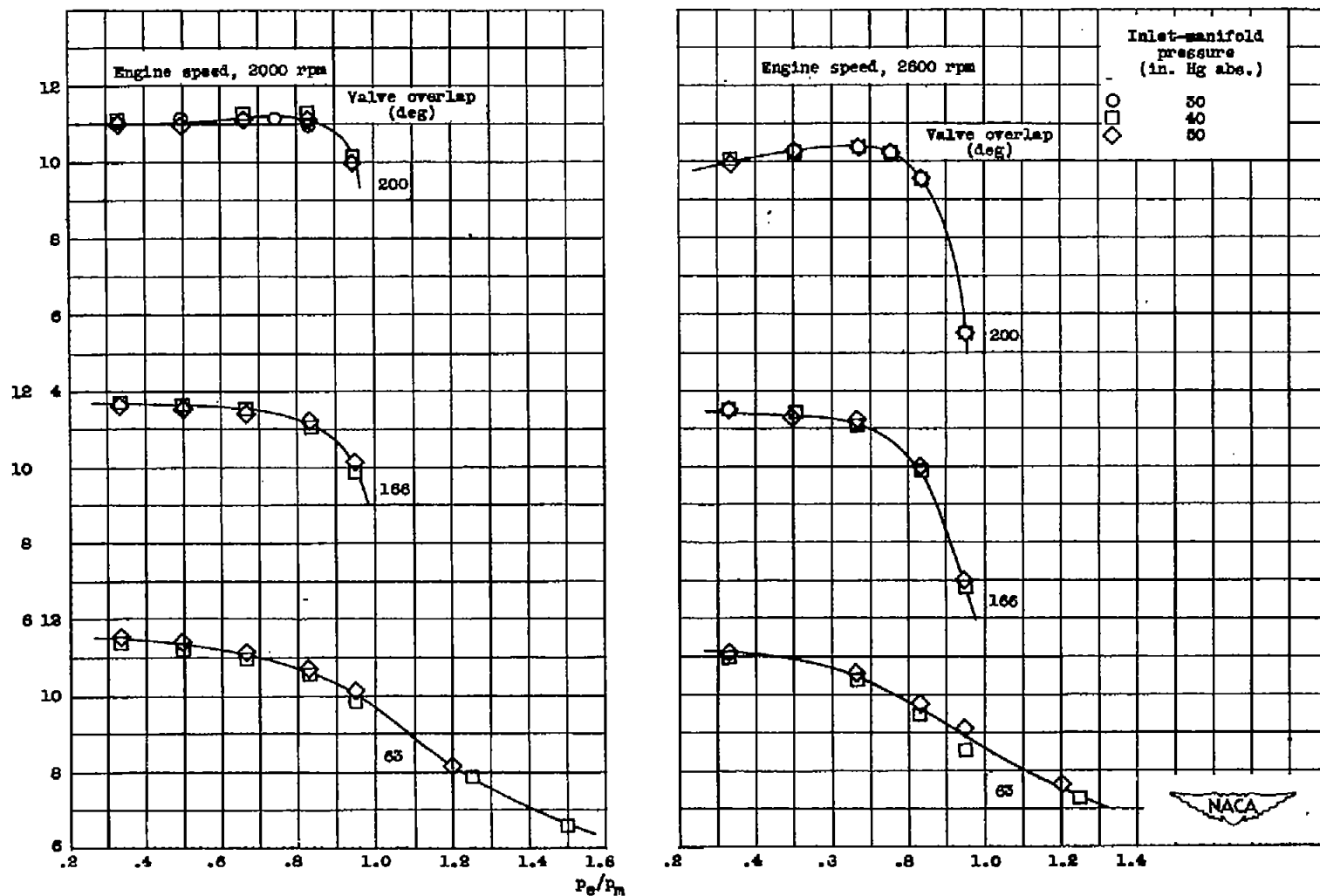
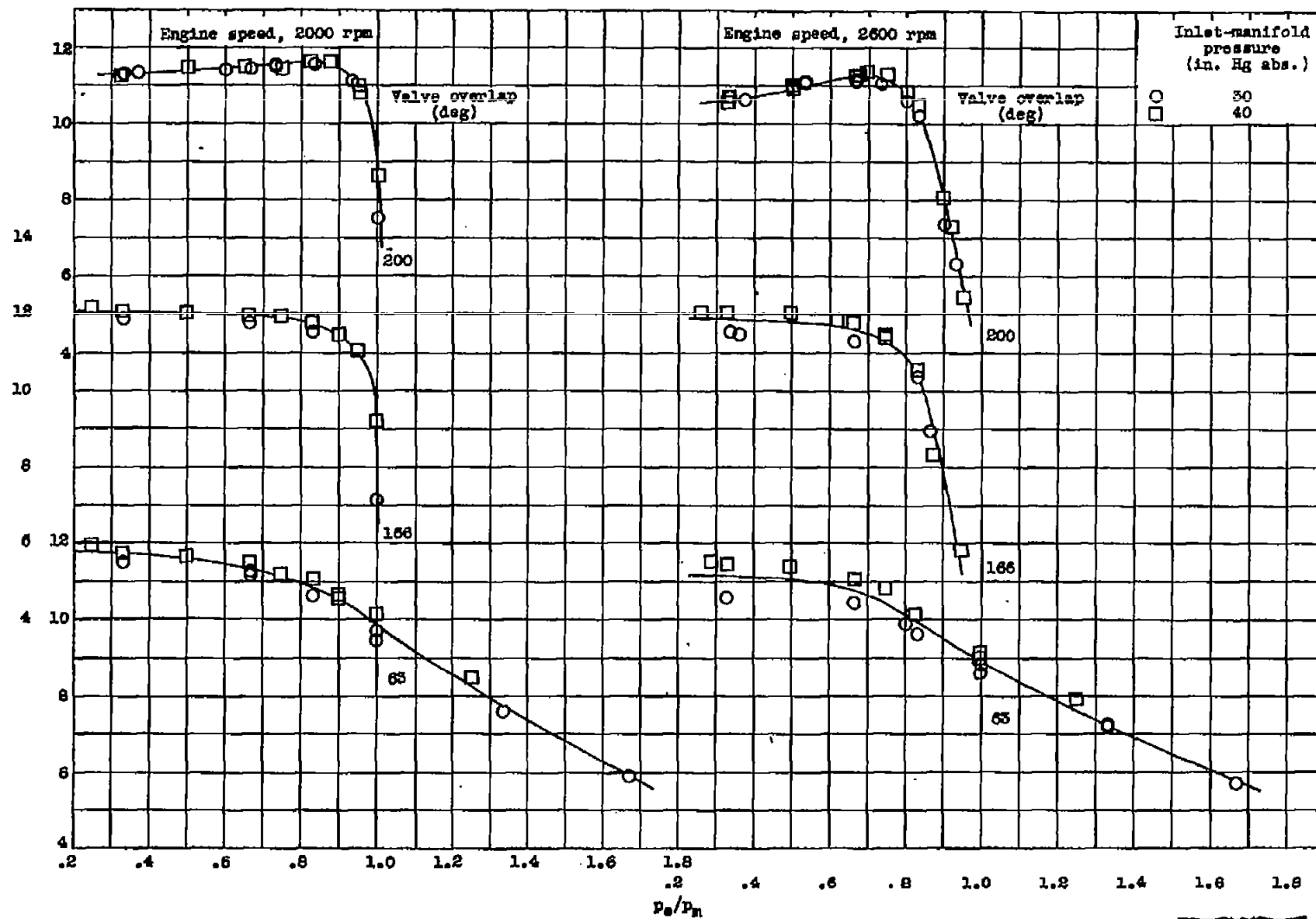
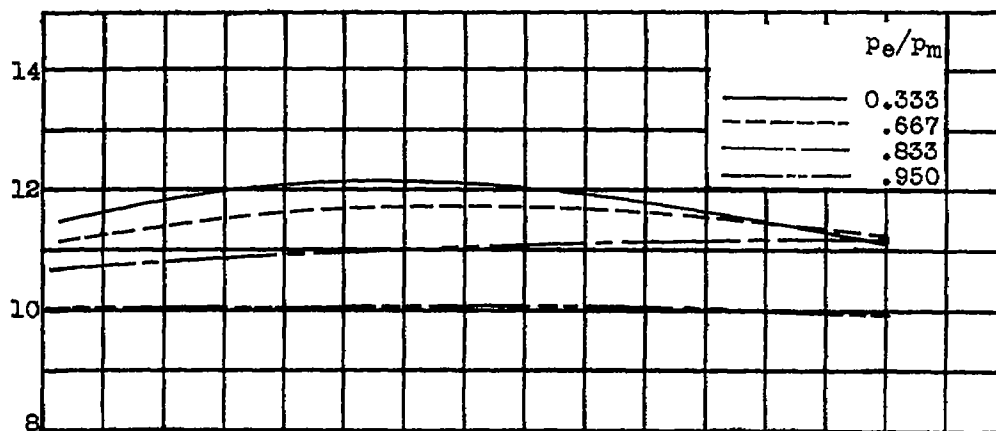


Figure 3. - Variation of ϕ with P_c/P_m . Over-all fuel-air ratio, 0.065.

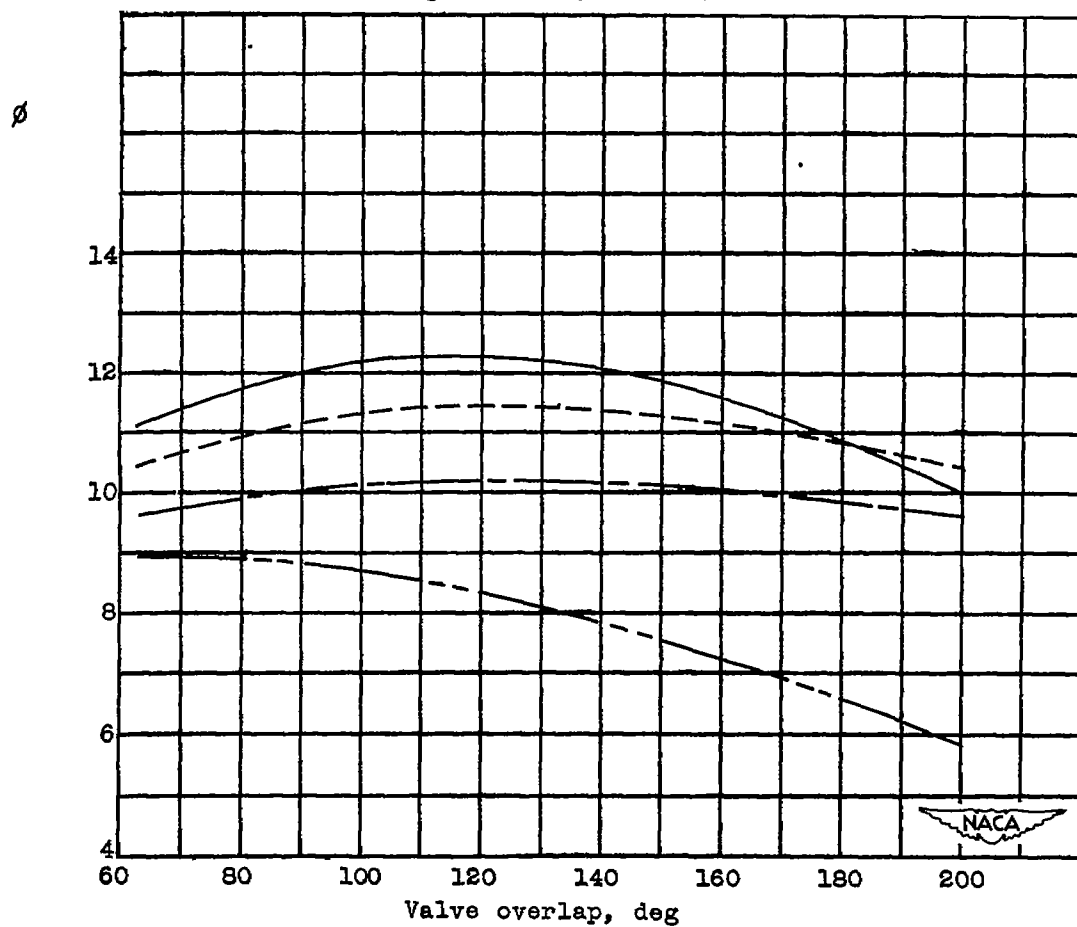


(b) Compression ratio, 7.6.

Figure 3. - Concluded. Variation of ϕ with p_c/p_m . Over-all fuel-air ratio, 0.065.

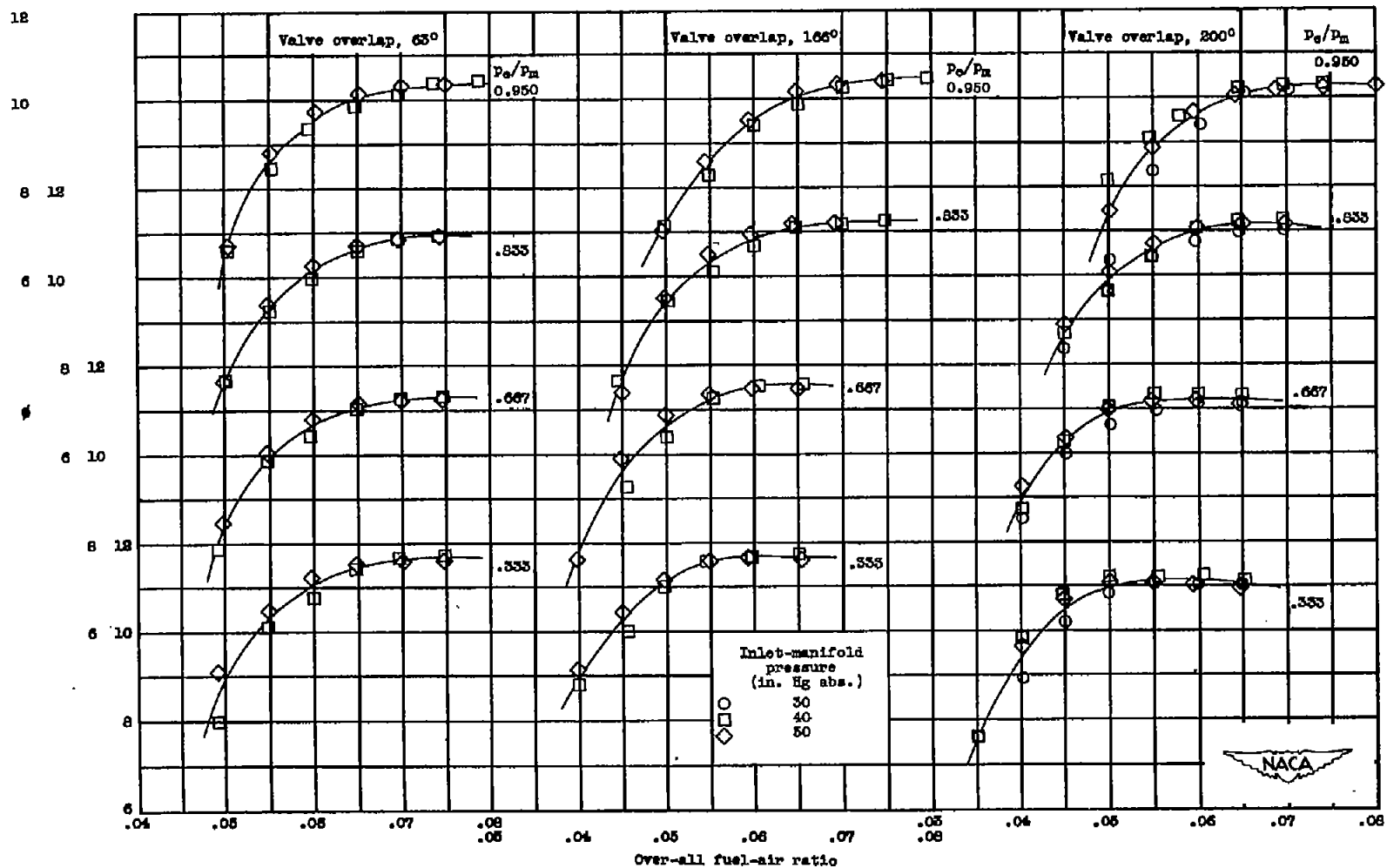


(a) Engine speed, 2000 rpm.

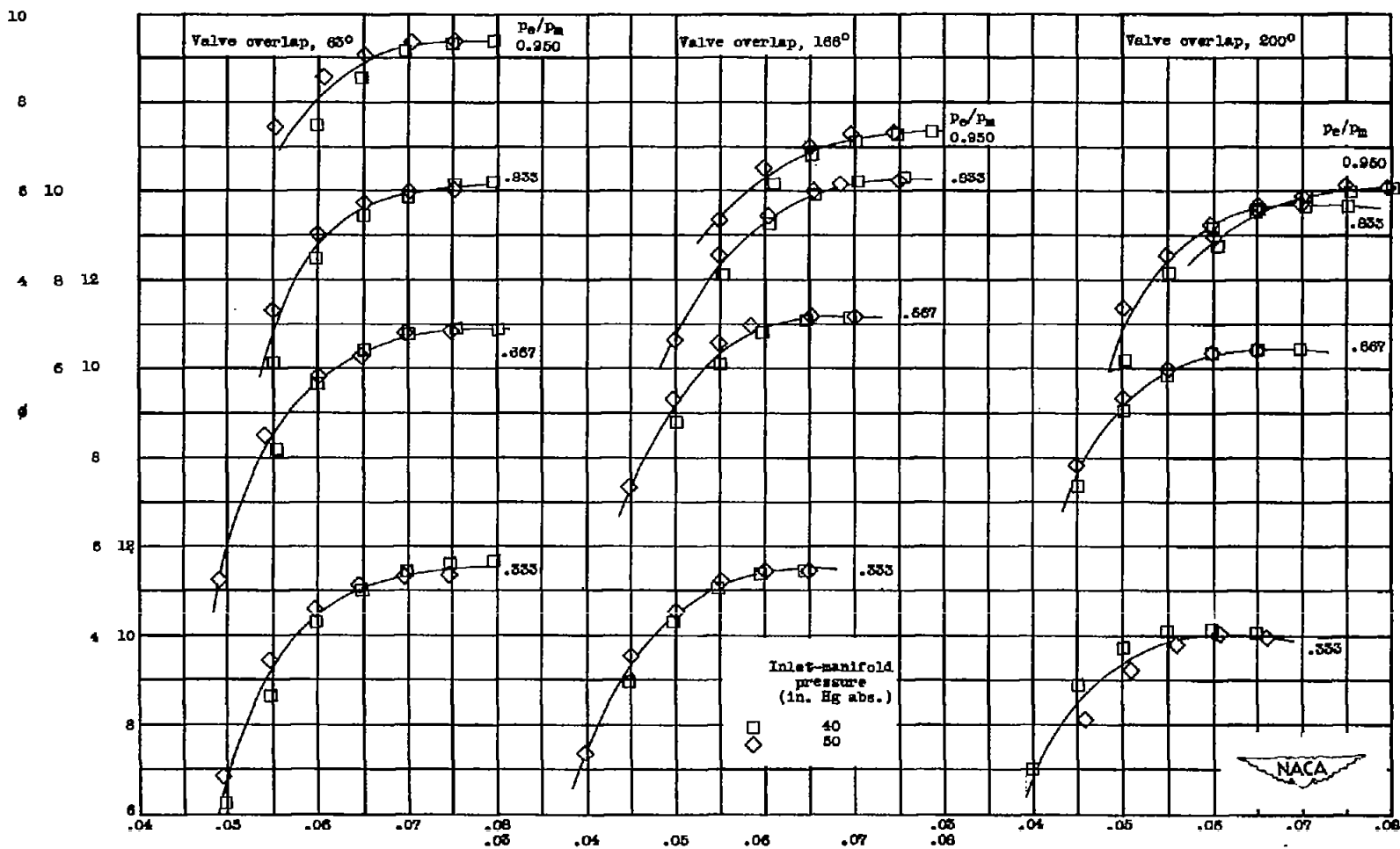


(b) Engine speed, 2600 rpm.

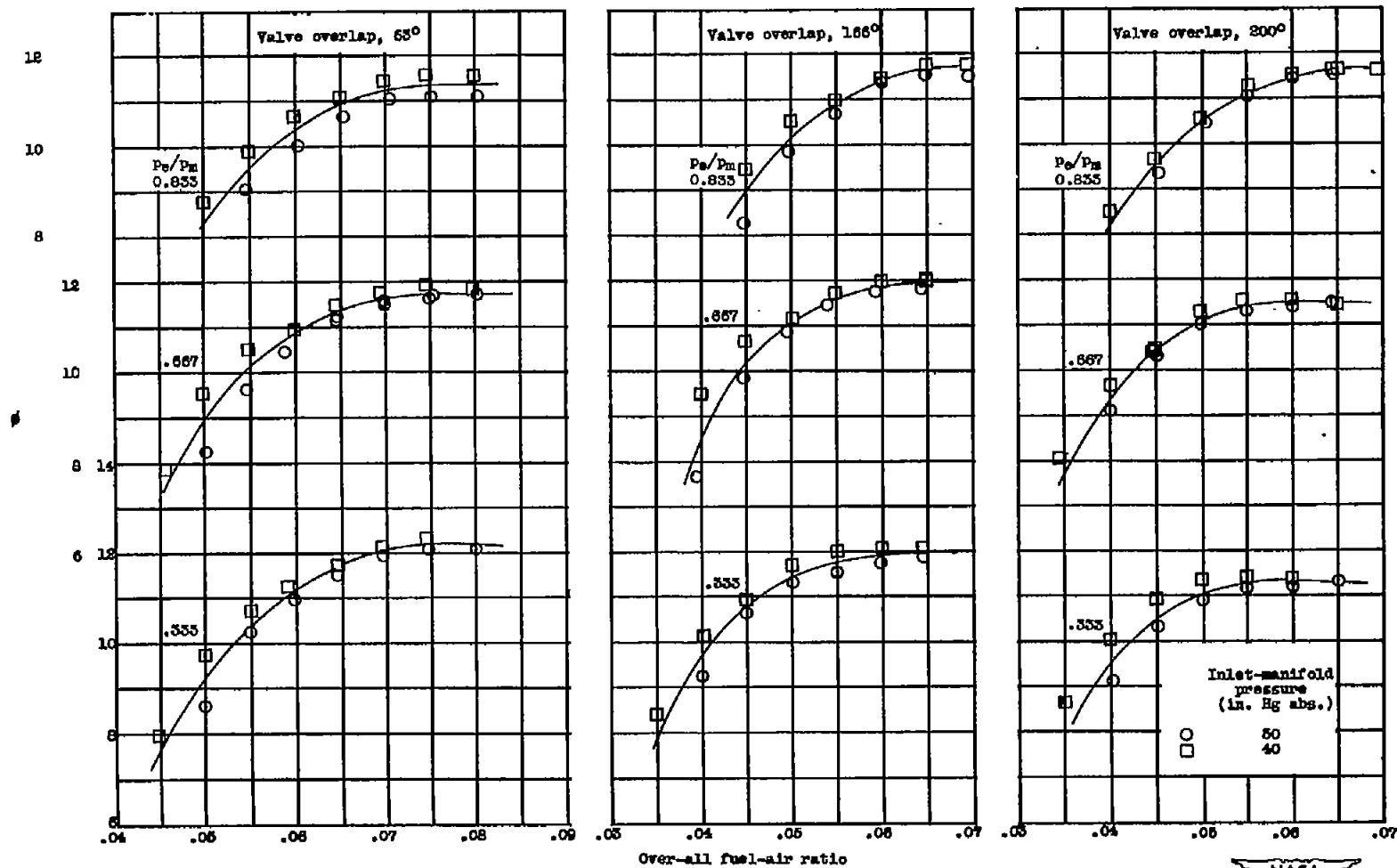
Figure 4. - Variation of ϕ with valve overlap. Over-all fuel-air ratio, 0.065; compression ratio, 5.5.



(a) Compression ratio, 8.5; engine speed, 2000 rpm.
 Figure 5. - Variation of ϕ with over-all fuel-air ratio.

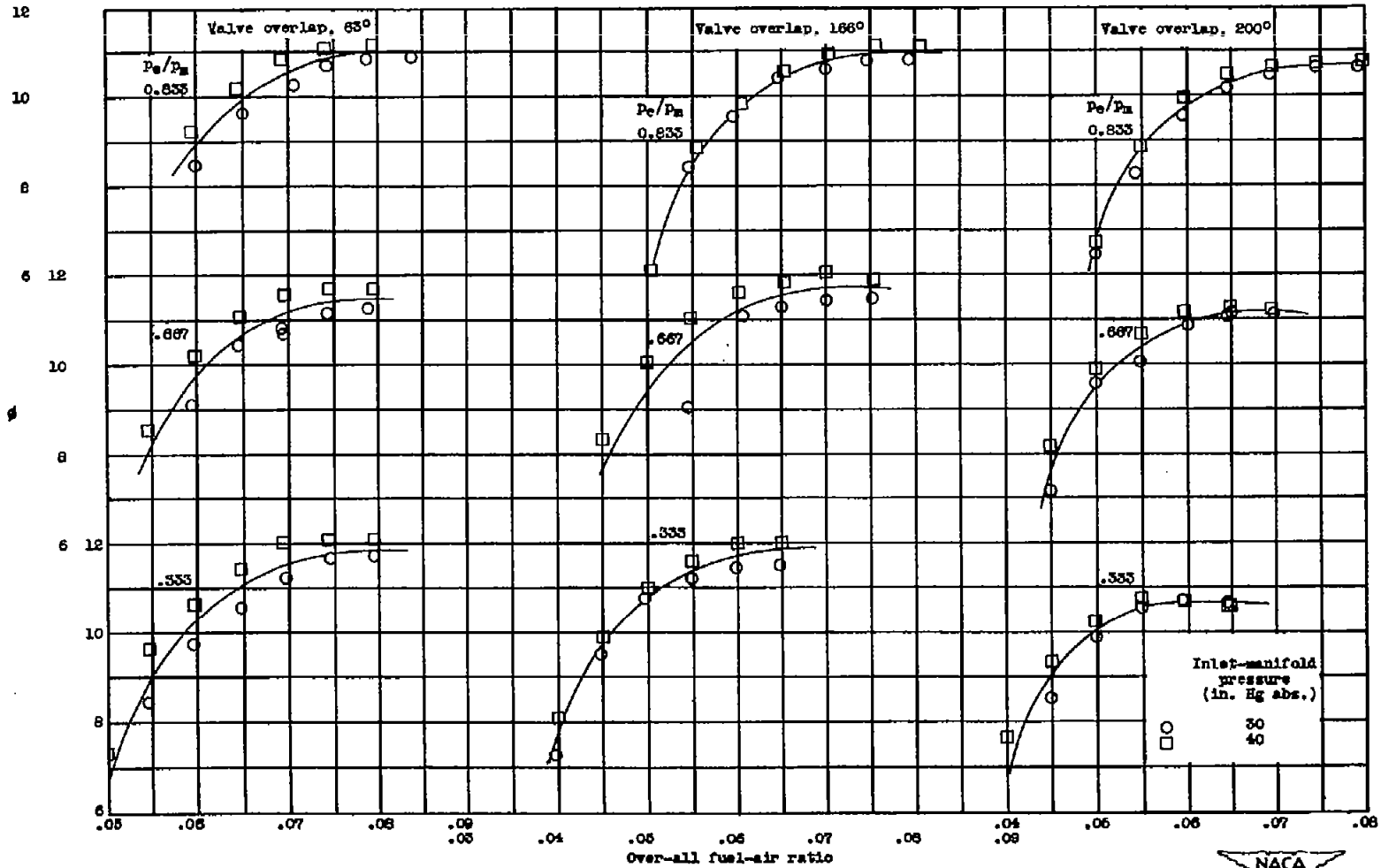


(b) Compression ratio, 5.6; engine speed, 2500 rpm.
Figure 5. - Continued. Variation of C_d with over-all fuel-air ratio.

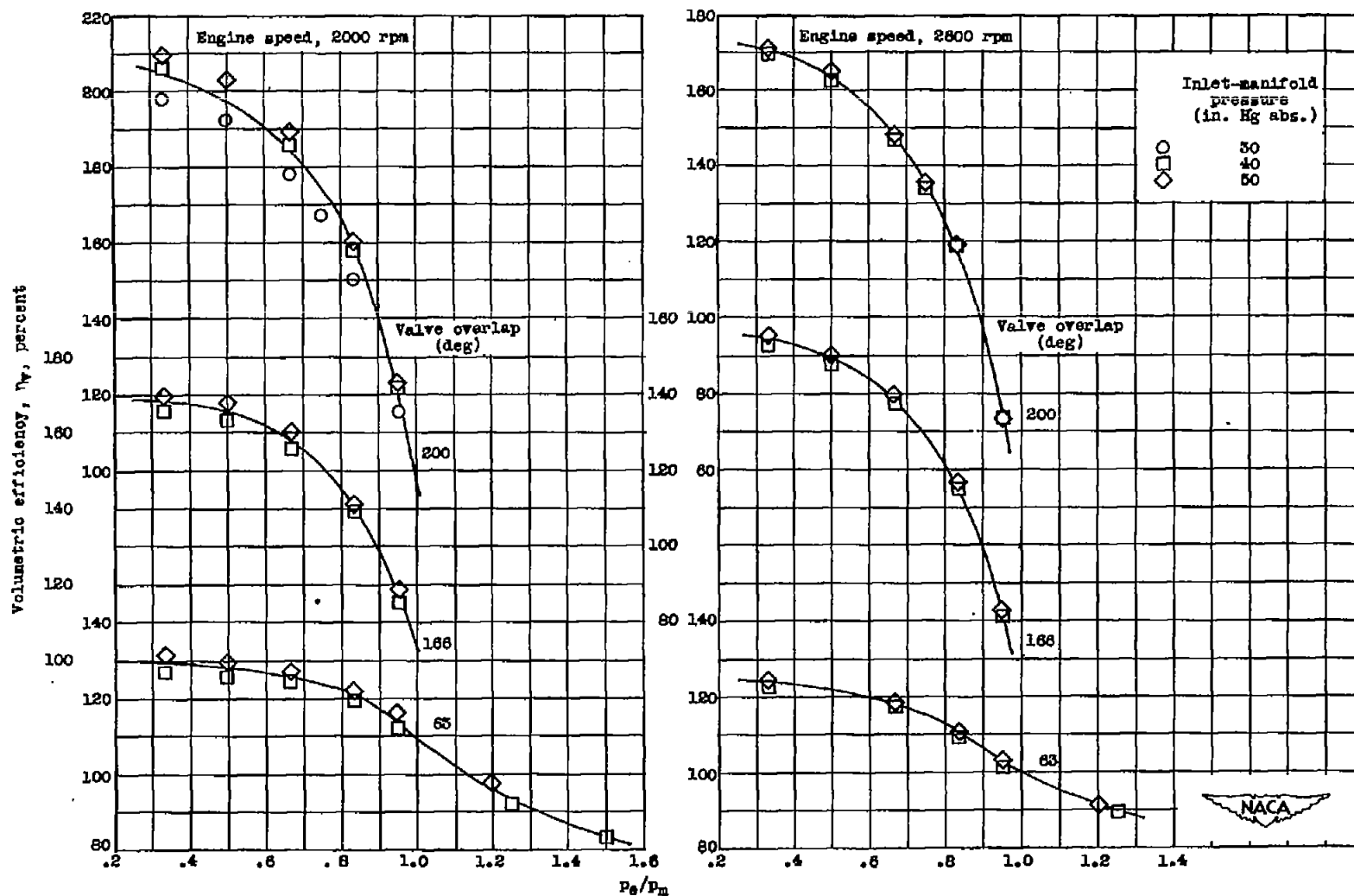


(c) Compression ratio, 7.6; engine speed, 2000 rpm.
Figure 5. - Continued. Variation of ϕ with over-all fuel-air ratio.





(1) Compression ratio, 7.6; engine speed, 2600 rpm.
Figure 5. - Concluded. Variation of ϕ with over-all fuel-air ratio.



(a) Compression ratio, 5.5.

Figure 6. - Variation of volumetric efficiency η_v with P_c/P_m . Over-all fuel-air ratio, 0.035.

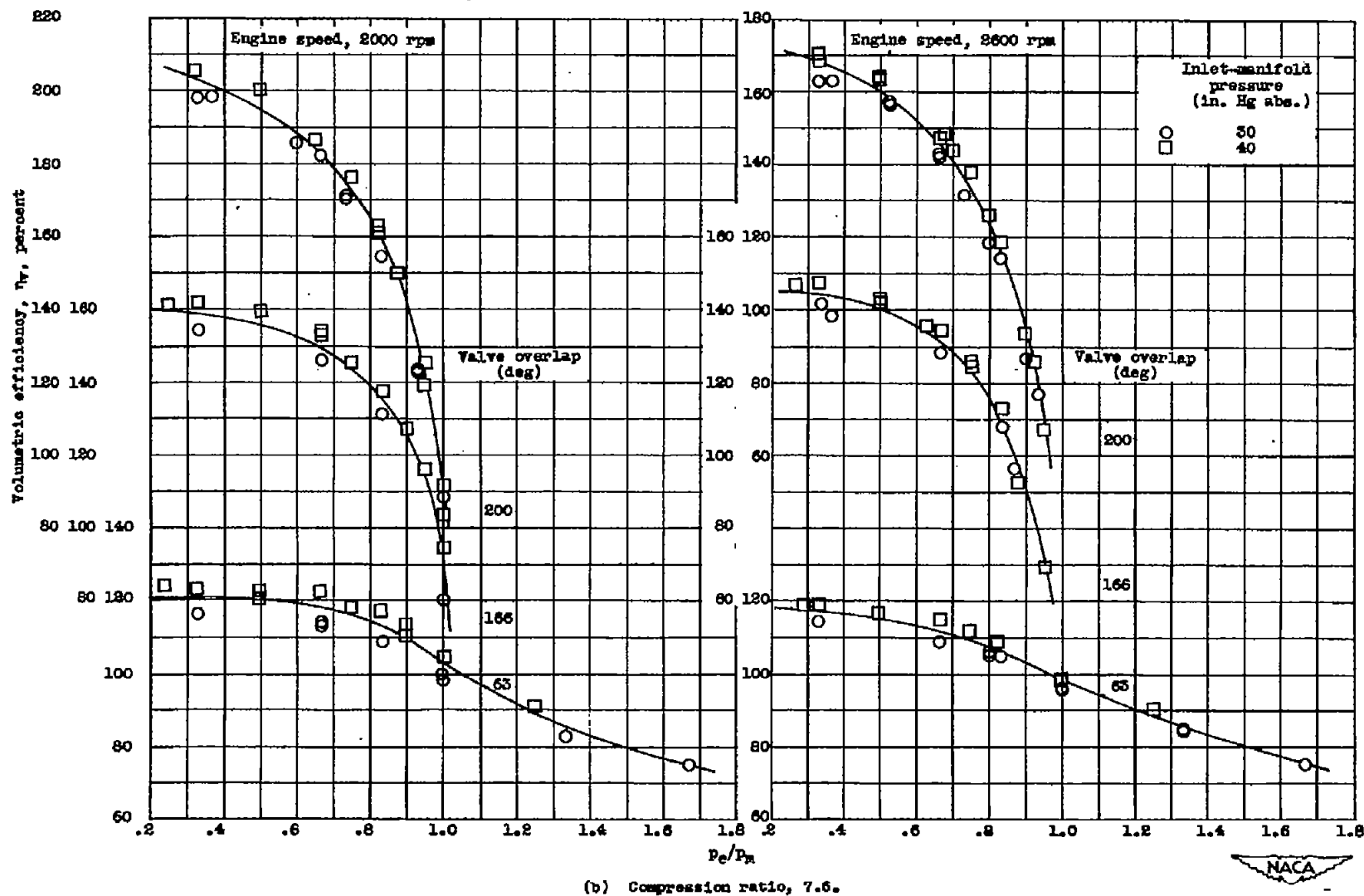


Figure 6. - Concluded. Variation of volumetric efficiency η_v with P_e/P_m . Over-all fuel-air ratio, 0.065.

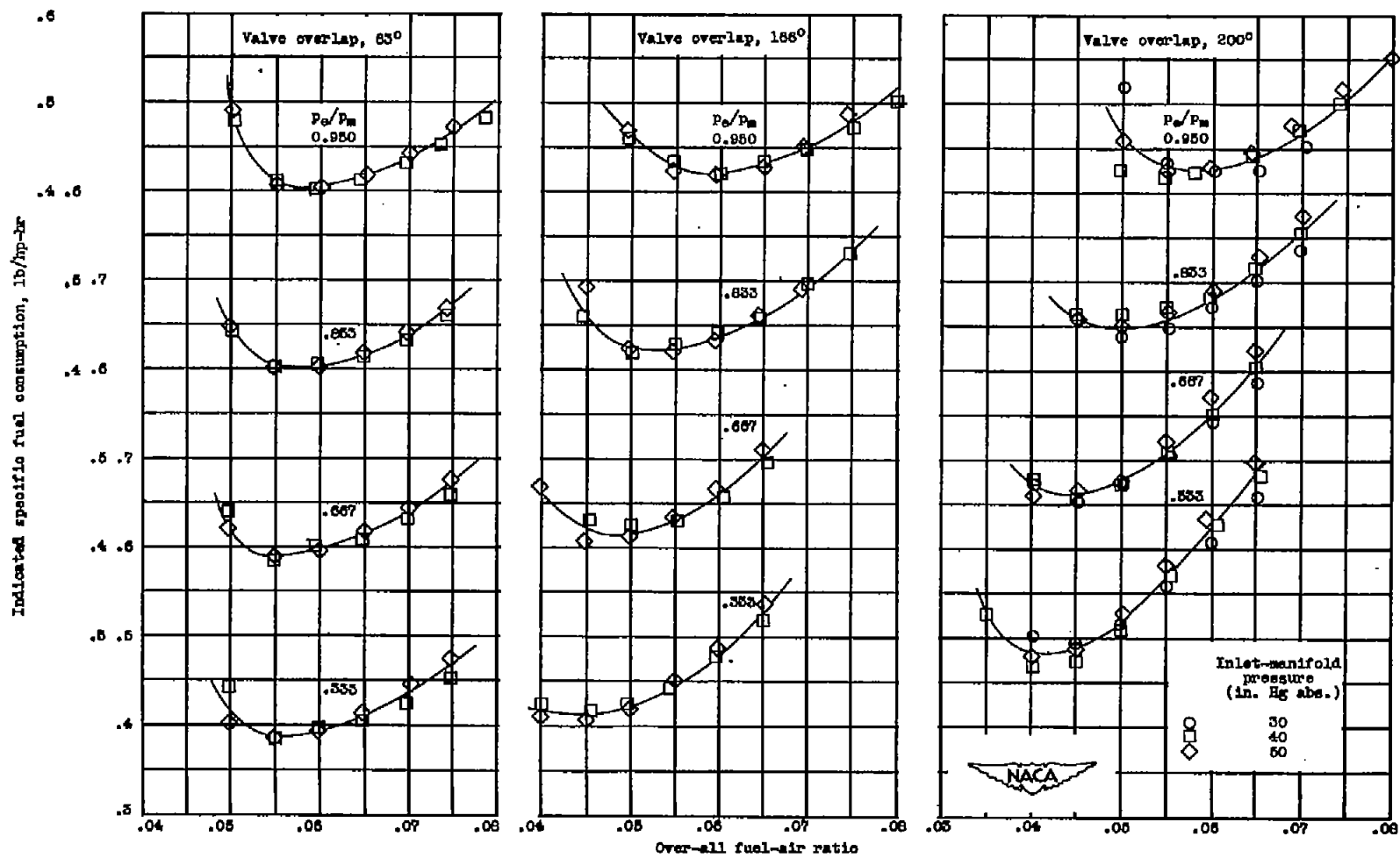


Figure 7. - Variation of indicated specific fuel consumption with over-all fuel-air ratio.

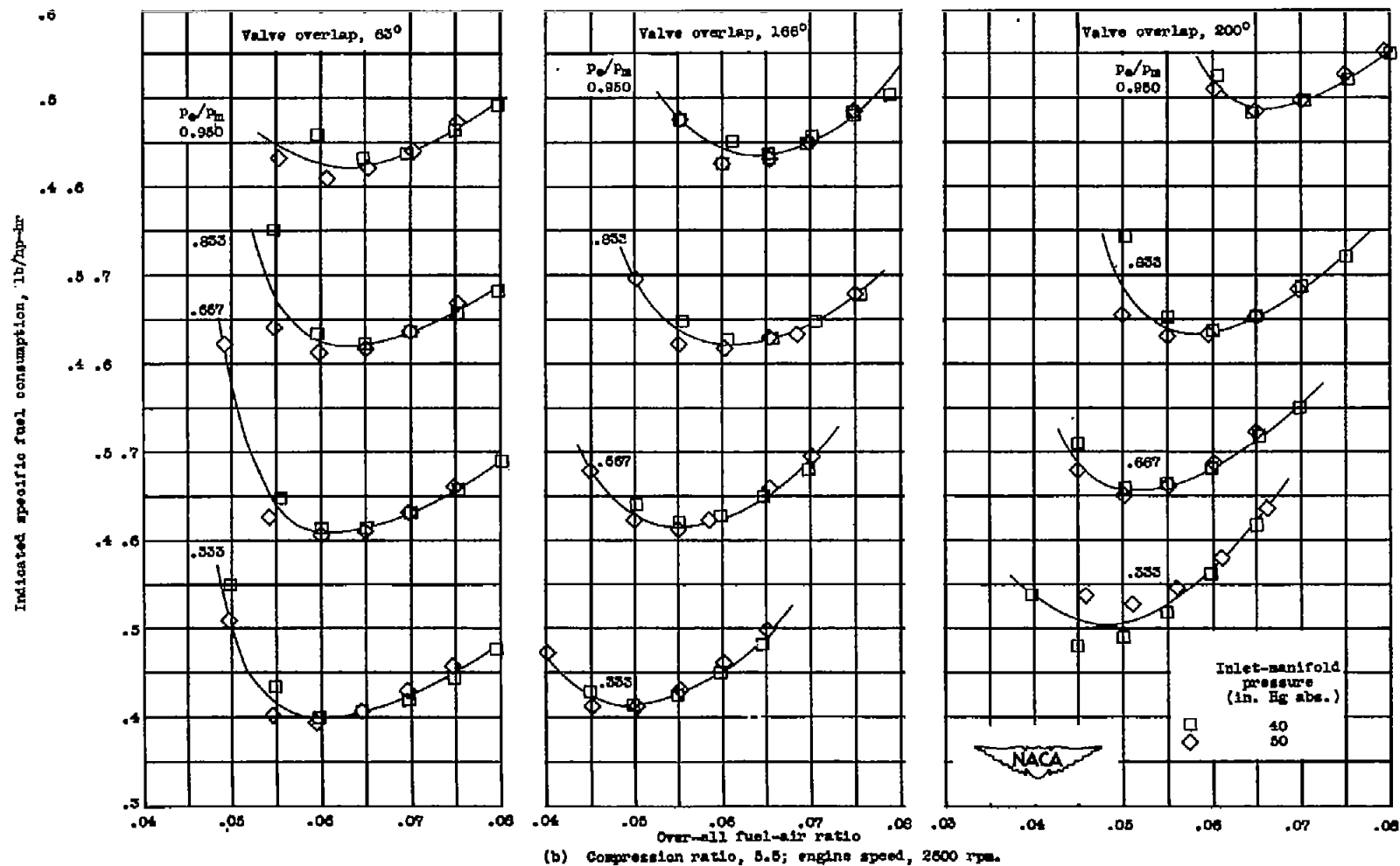
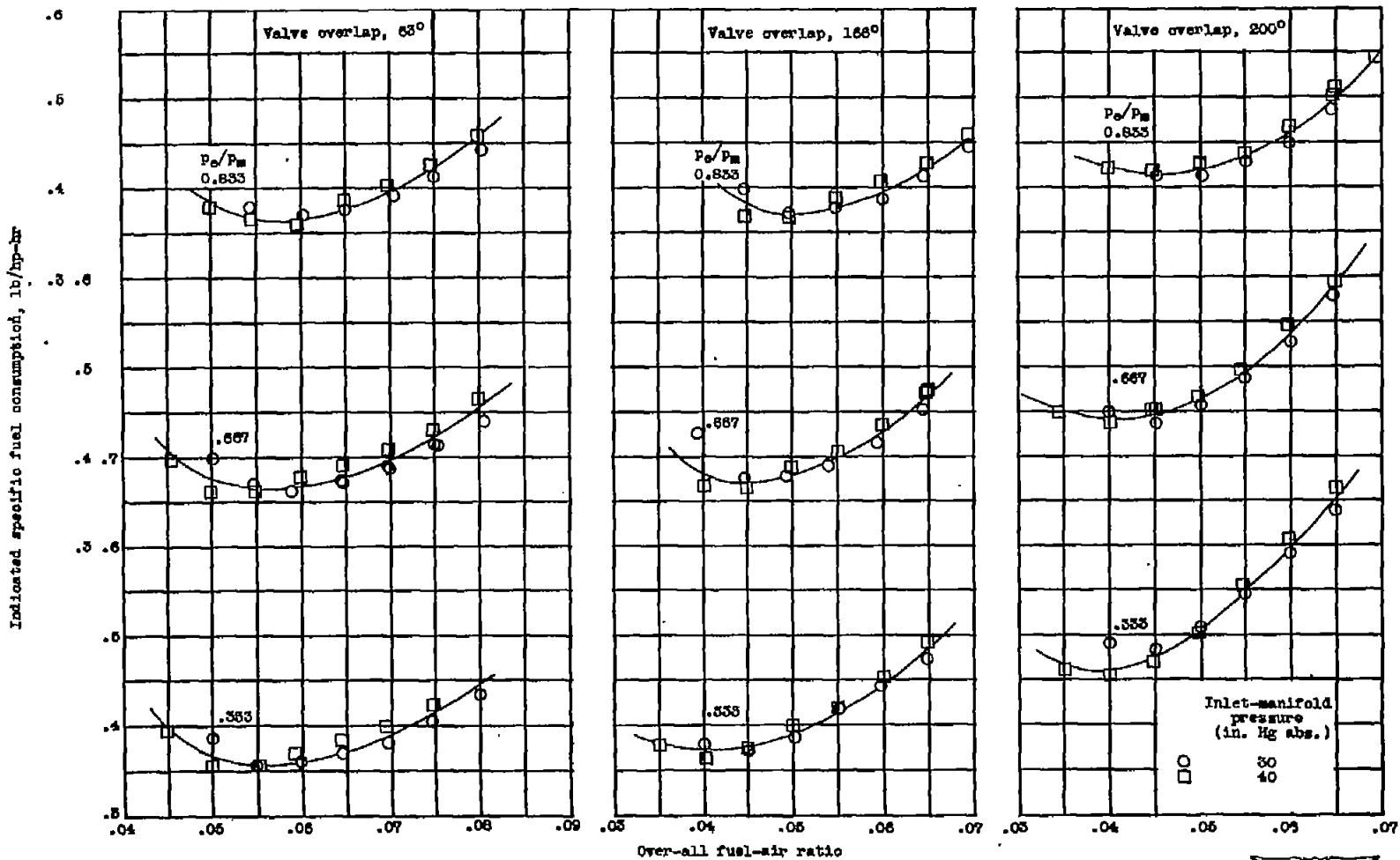
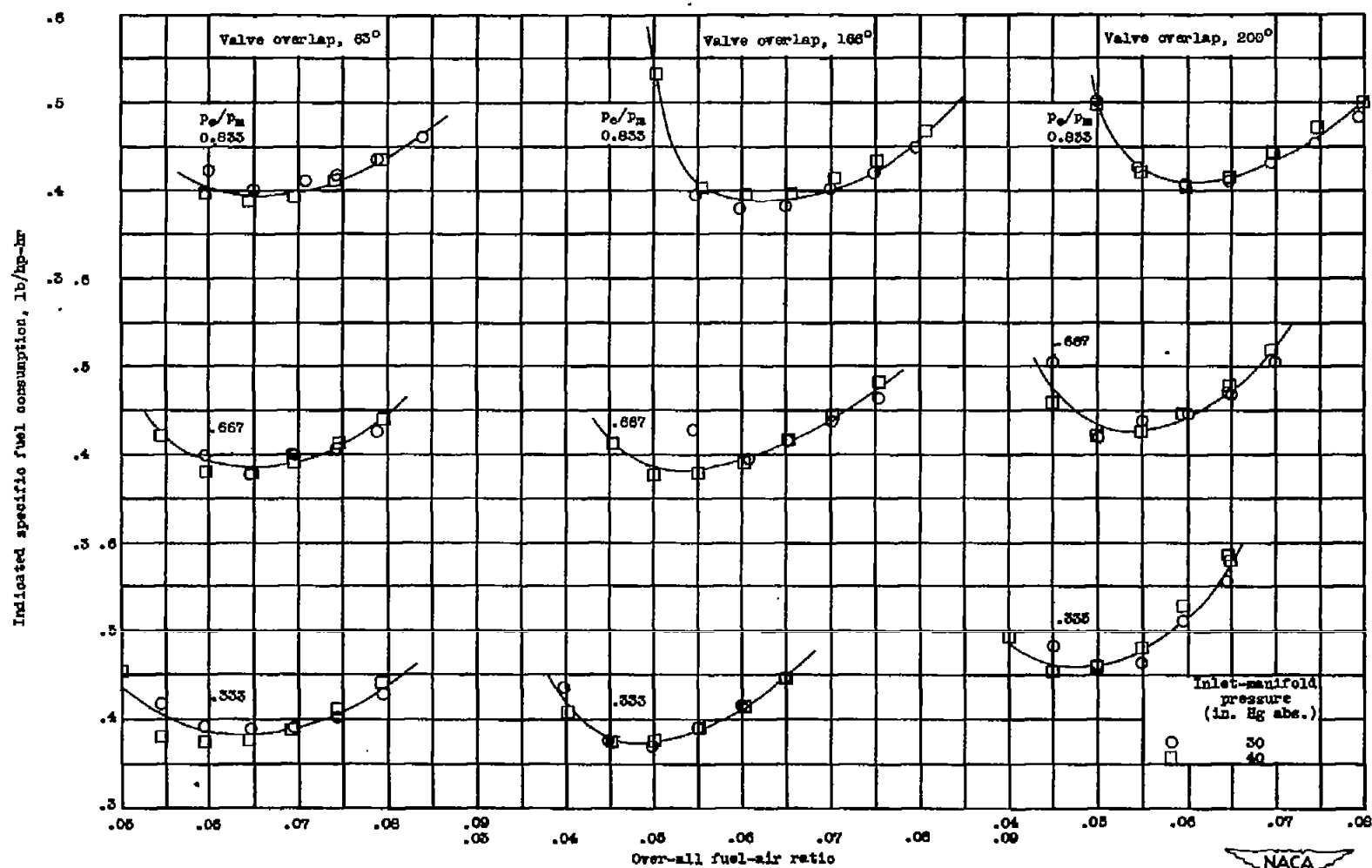


Figure 7. - Continued. Variation of indicated specific fuel consumption with over-all fuel-air ratio.



(c) Compression ratio, 7.6; engine speed, 2000 rpm.
Figure 7. - Continued. Variation of indicated specific fuel consumption with over-all fuel-air ratio.





(d) Compression ratio, 7.6; engine speed, 2600 rpm.
Figure 7. - Concluded. Variation of indicated specific fuel consumption with over-all fuel-air ratio.

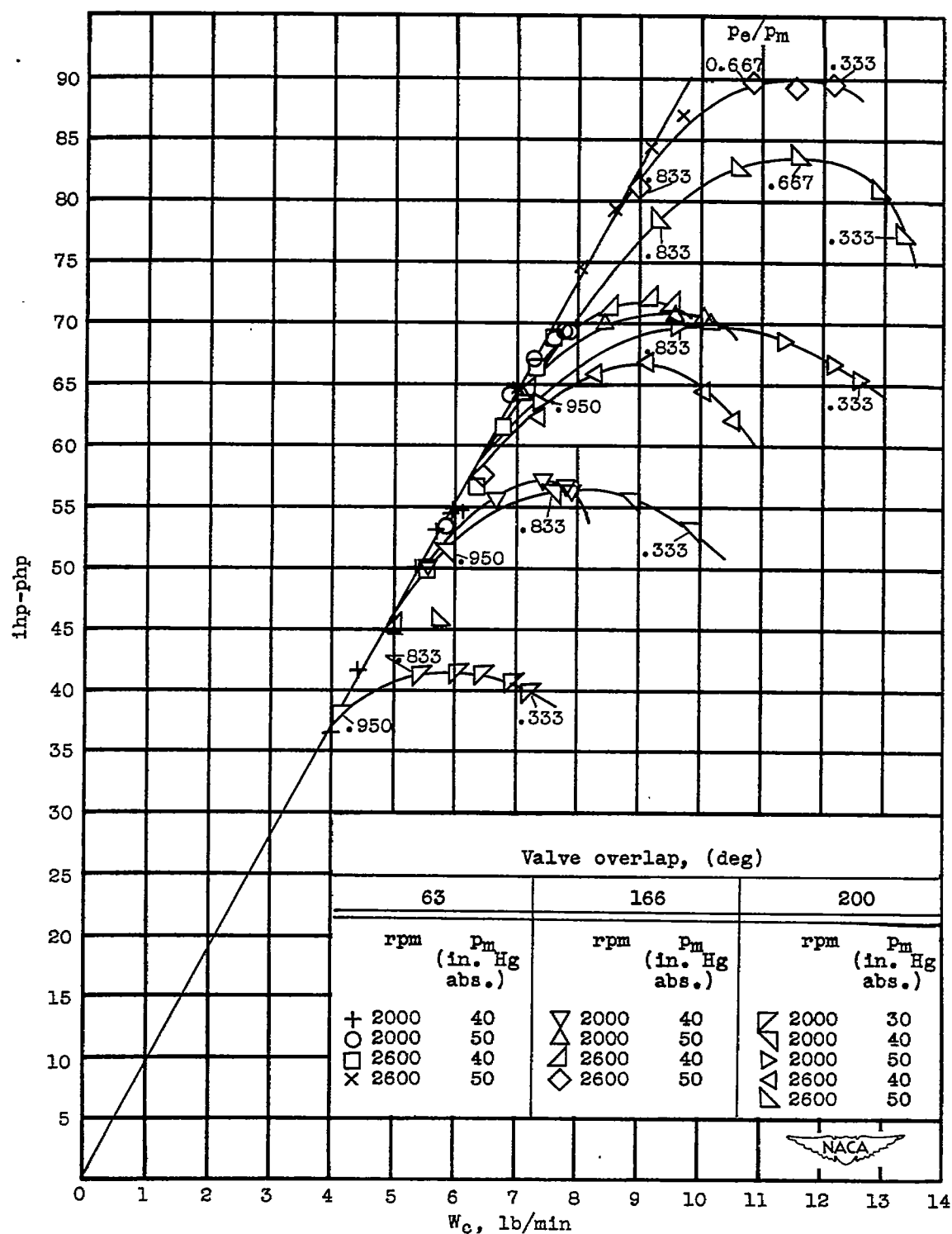
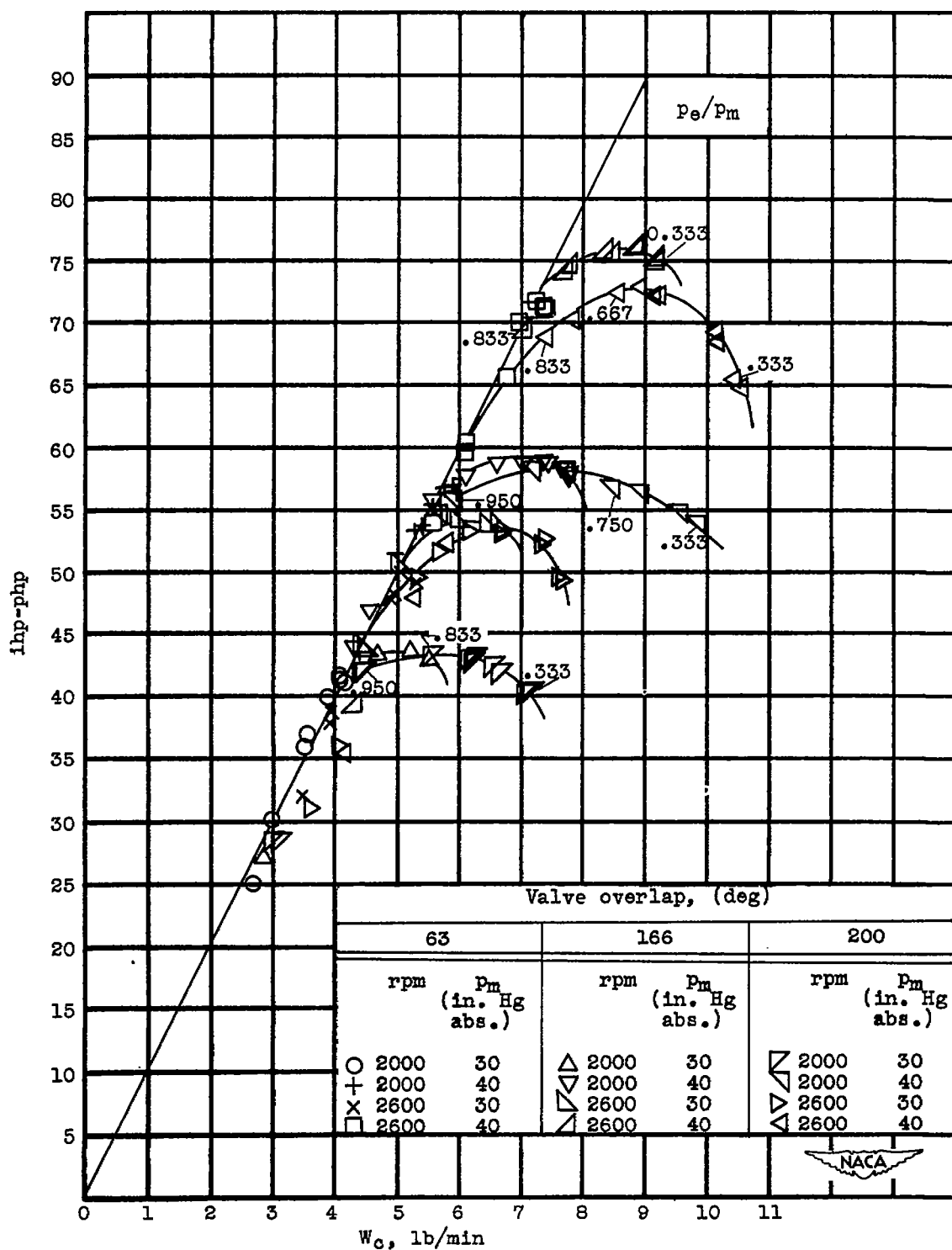


Figure 8. - Variation of indicated horsepower minus pumping horsepower with charge-air flow. Over-all fuel-air ratio, 0.065.



(b) Compression ratio, 7.6.

Figure 8. - Concluded. Variation of indicated horsepower minus pumping horsepower with charge-air flow. Over-all fuel-air ratio, 0.065.

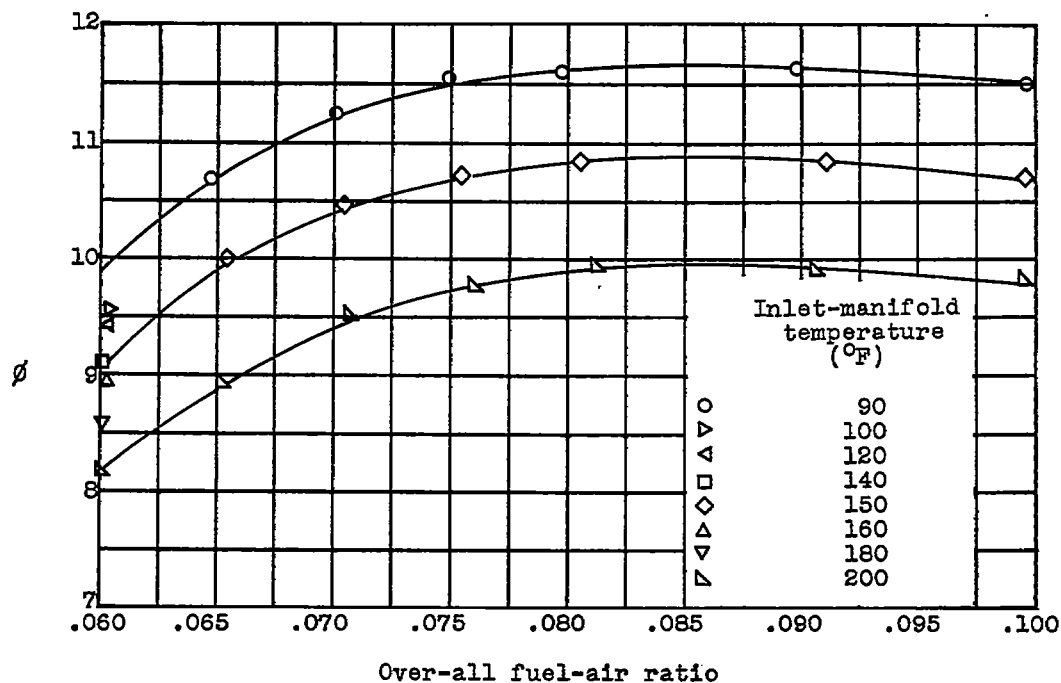


Figure 9. - Variation of ϕ with over-all fuel-air ratio. Compression ratio, 7.6; valve overlap, 63°; engine speed, 2000 rpm; inlet-manifold pressure, 30 inches mercury absolute; exhaust pressure, 30 inches mercury absolute.

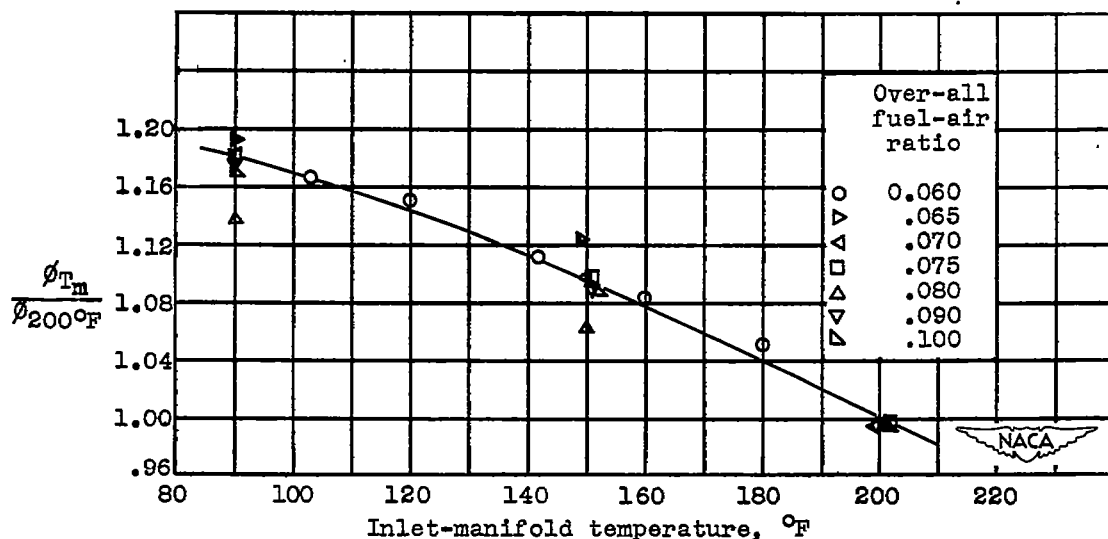


Figure 10. - Variation of ϕ at any inlet-manifold temperature to ϕ at inlet-manifold temperature of 200° F with inlet-manifold temperature. Compression ratio, 7.6; valve overlap, 63°; engine speed, 2000 rpm; inlet-manifold pressure, 30 inches mercury absolute; exhaust pressure, 30 inches mercury absolute.

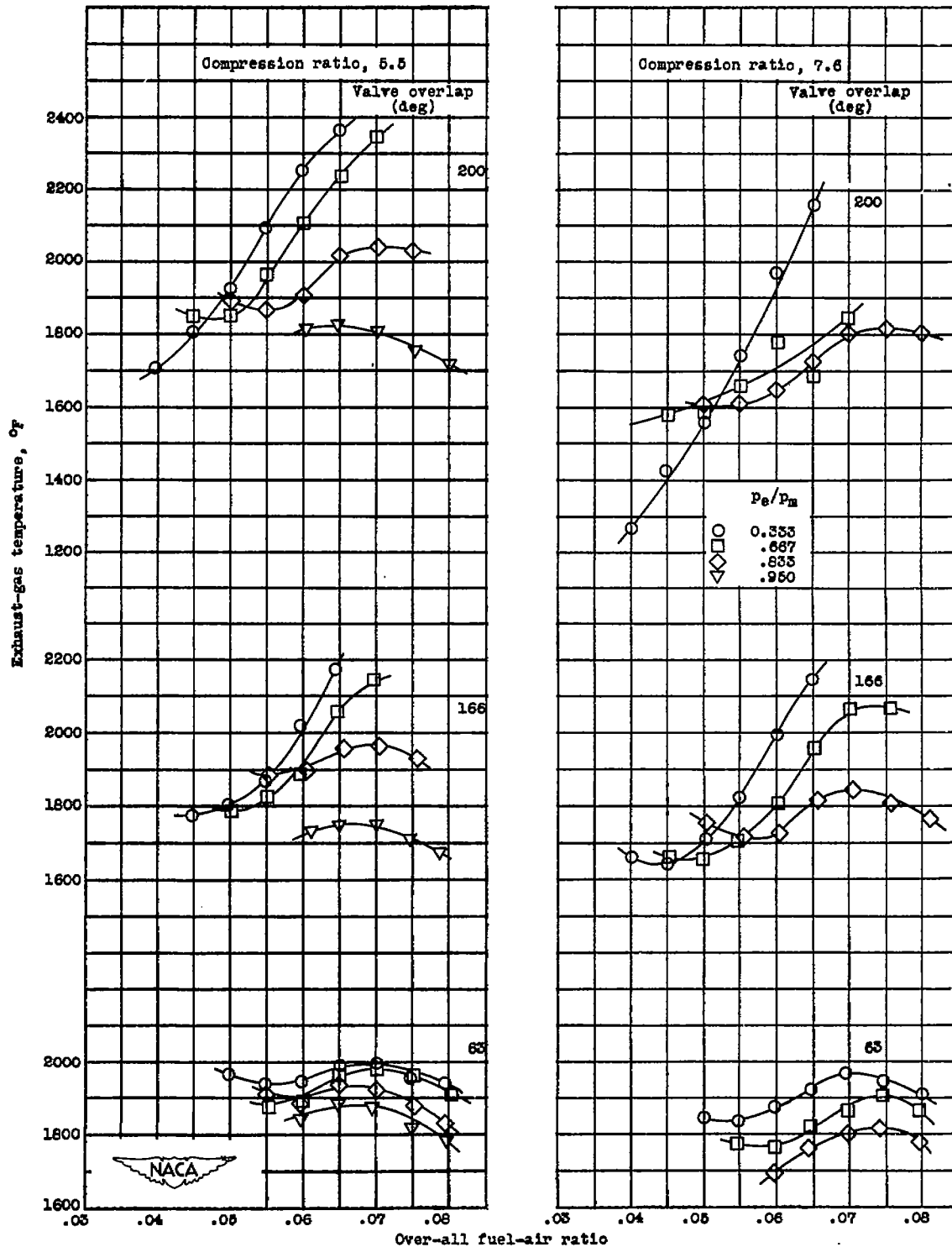


Figure 11. - Variation of exhaust-gas temperature with over-all fuel-air ratio. Engine speed, 2600 rpm; inlet-manifold pressure, 40 inches mercury absolute.

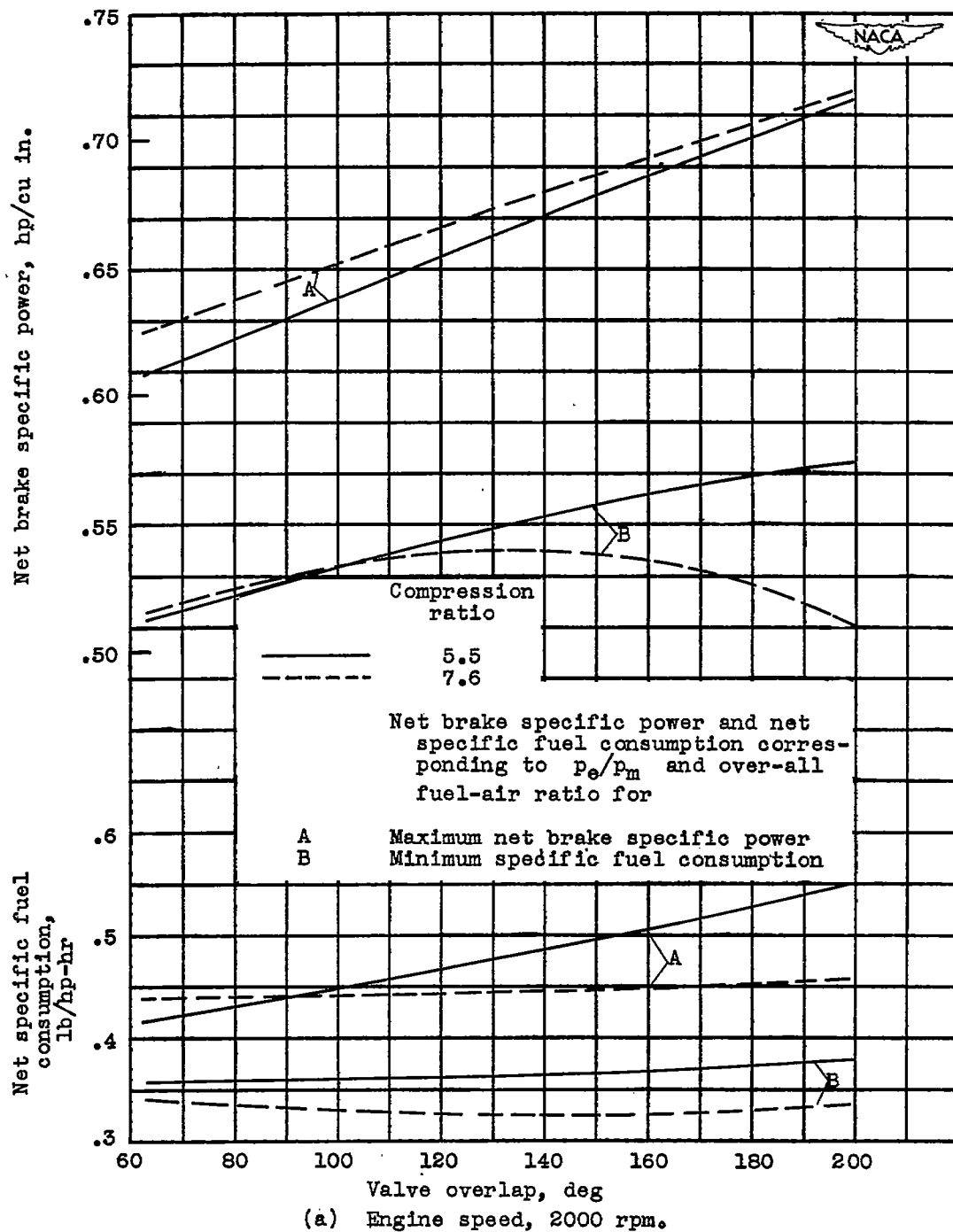


Figure 12. - Variation of net brake specific power and net specific fuel consumption with valve overlap for compound power plant. Altitude, 30,000 feet; inlet-manifold pressure, 40 inches mercury absolute.

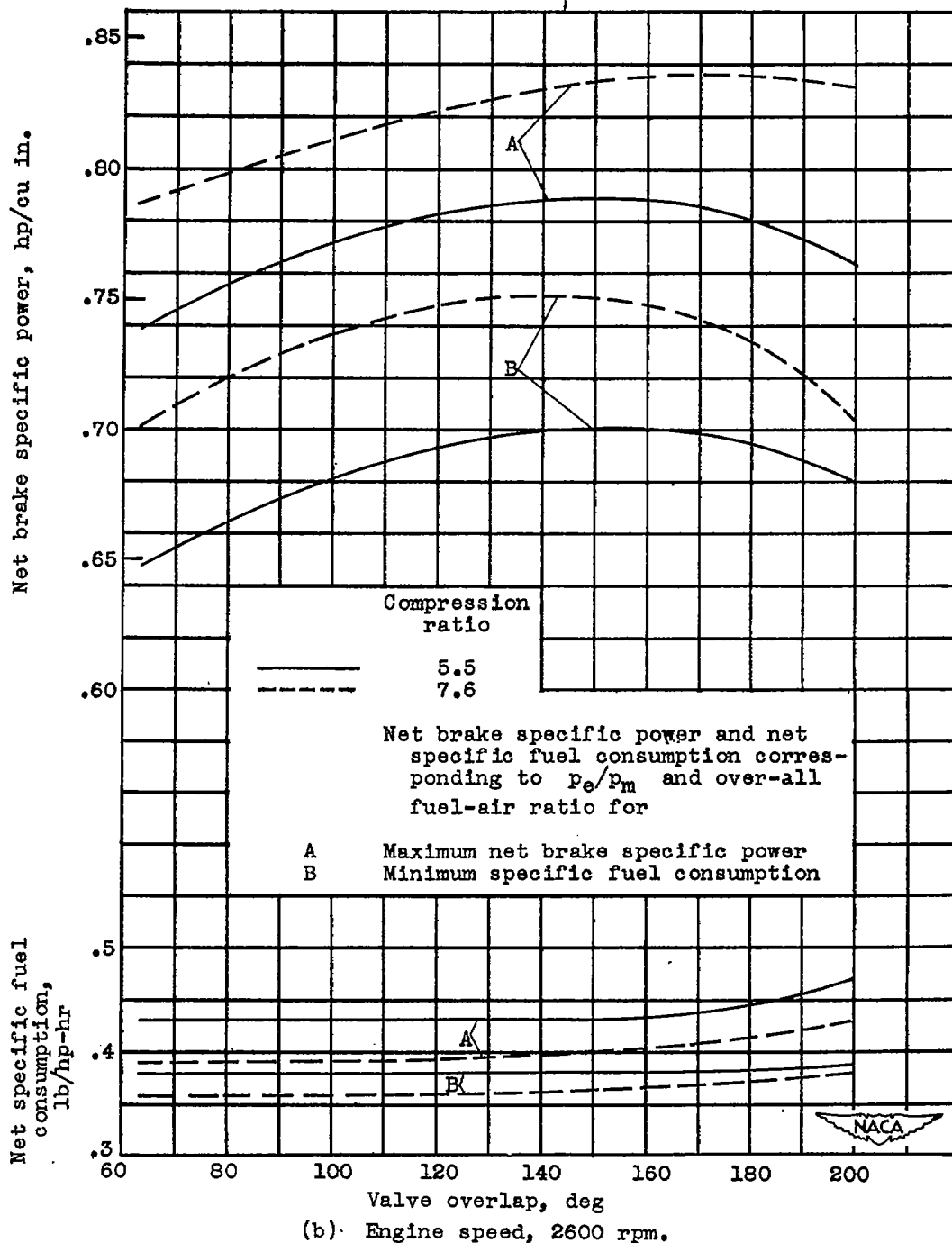


Figure 12. - Concluded. Variation of net brake specific power and net specific fuel consumption with valve overlap for compound power plant. Altitude, 30,000 feet; inlet-manifold pressure, 40 inches mercury absolute.